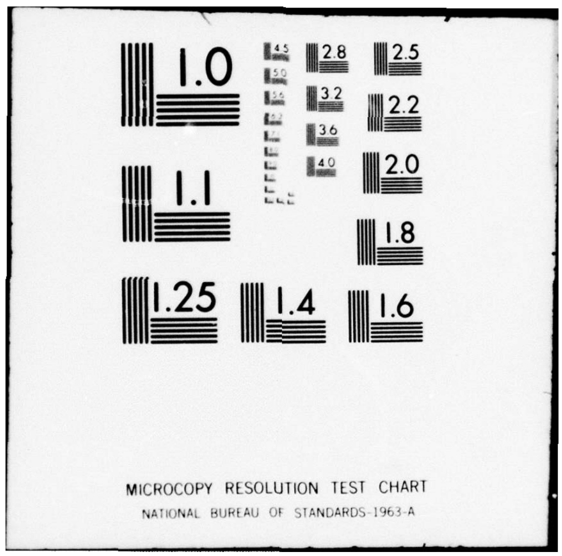
AD-A058 626 DUBIN-BLOOME ASSOCIATES NEW YORK F/G 13/1
SOLAR ASSISTED HEAT PUMP STUDY FOR HEATING OF MILITARY FACILITI--ETC(U)
JUL 78 F L BEASON, L W STROTHER F08635-76-C-0276
UNCLASSIFIED AFCEC-TR-78-6 NL

1 OF 3
AD
A058626


MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

AFCEC-TR-78-6

SOLAR ASSISTED HEAT PUMP STUDY FOR HEATING OF MILITARY FACILITIES

DUBIN-BLOOME ASSOCIATES, P.C.,
ENERGY MANAGEMENT CONSULTANTS
NEW YORK, NEW YORK
WEST HARTFORD, CONNECTICUT

JULY 1978

Final Report For Period July 1976-December 1977

Approved for Public Release; Distribution Unlimited.

AIR FORCE CIVIL ENGINEERING CENTER
(AIR FORCE ENGINEERING AND SERVICES CENTER)
TYNDALL AIR FORCE BASE
FLORIDA 32403

UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

19 **REPORT DOCUMENTATION PAGE**

READ INSTRUCTIONS BEFORE COMPLETING FORM

1. REPORT NUMBER: 18 AFCEC-TR-78-6	2. GOVT ACCESSION NO.
3. RECIPIENT'S CATALOG NUMBER: 9 rept.	
4. TITLE (and Subtitle): 6 Solar Assisted Heat Pump Study For Heating Of Military Facilities	5. TYPE OF REPORT & PERIOD COVERED: Final Jul 76 — Nov 77
	6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s): 10 Frederick L. Beason / Larry W. Strother, AFCEC/DEM, Tyndall AFB, FL 32403	8. CONTRACT OR GRANT NUMBER(s): 15 FØ8635-76C-0276
9. PERFORMING ORGANIZATION NAME AND ADDRESS: Dubin-Bloome Associates ~~P.C.; Energy Management Consultants~~; New York~~, New York~~ West Hartford, Connecticut	10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS: P-721 Purchase Order ØØDEMEØ3
11. CONTROLLING OFFICE NAME AND ADDRESS: Air Force Civil Engineering Center (AFESC) Tyndall Air Force Base, Florida 32403	12. REPORT DATE: 11 Jul 78
	13. NUMBER OF PAGES
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office): HQ USAF/LEEN Washington DC 20330 — 12 2Ø2p.	15. SECURITY CLASS. (of this report): UNCLASSIFIED
	15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for Public Release; Distribution Unlimited

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

Available in DDC

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Heat Pumps
Solar Energy
Solar Heating
Solar Preheating
Energy Conservation
Retrofit

20. ABSTRACT (Continue on reverse side if necessary and identify by block number) This study identified 21 generic solar assisted heat pump systems and subjectively evaluated them. The six most promising systems were evaluated in further detail. A complete objective analysis of the two most promising systems was then made to determine which could be most economical to install in a family housing unit at Little Rock Air Force Base, Arkansas. The system chosen was a solar hot water heating system in parallel with a water source heat pump. Preliminary drawings intergrating this system into a family housing unit at Little Rock AFB were developed. The system selected had a 27-32 year pay back.

1B

DD FORM 1 JAN 73 1473 EDITION OF 1 NOV 65 IS OBSOLETE

UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

S/C 392972

PREFACE

This report discusses the applicability of using solar energy to assist and improve the efficiency of heat pumps used in Air Force family housing units, particularly those at Little Rock AFB, AK. It combines four reports prepared as the results of a study performed by Dubin-Bloome Associates for the Air Force Civil Engineering Center (AFCEC). This study was conducted for Headquarters Air Force, Directorate of Engineering & Services, Housing Division.

Dubin-Bloome Associates' principal advisors were Fred S. Dubin and H. Robert Sparkes. Their project managers were Barry O. Symmonds and Philip Fine. The AFCEC project officer was Freddie L. Beason. Larry W. Strother and Freddie L. Beason combined the four reports developed by Dubin-Bloome Associates into this technical report.

This report has been reviewed by the Information Officer (OI) and is releasable to the National Technical Information Service (NTIS). At NTIS it will be available to the general public, including foreign nations.

This report has been reviewed and is approved for publication.

FREDDIE L BEASON, P.E. Energy Consultant	A STANLEY DALTON Director, Engr Materials
ROBERT E BRANDON Deputy Director	GEORGE E ELLIS, Col USAF Commander

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

SECTION I

INTRODUCTION

1.1 BACKGROUND

The scope of this study was to identify generic solar assisted heat pump systems which can be applied to any house regardless of geographic location or climate.

This study was divided into three phases. The intent of Phase I was to subjectively analyze solar-assisted heat pump systems and select the six most promising.

The second phase was to make a complete objective analysis of the two most promising systems selected from Phase I, including capital and operating costs to determine which would be more favorable for installation in houses at Little Rock Air Force Base, Arkansas. During this phase of the study, a suitable house at the Little Rock Air Force Base was selected.

Subsequent to the completion of Phase II, the original scope of work was amended to include the evaluation of a solar-assisted hybrid heat pump system at Little Rock AFB and to determine the suitability of this system for Scott AFB, Illinois.

The third phase of the study was to develop design drawings and specifications for the most promising system selected from Phase II and show how it fits into the selected house. This phase also includes:

a. Recommendations for retrofitting the house to improve its thermal properties to reduce both heating and cooling loads.

b. Cost estimates for the system installation and building retrofit.

c. Design calculations.

d. Evaluation of energy use and economics.

1.2. PHASE I

<u>Summary</u>.

Twenty-one candidate systems were selected and subjectively evaluated. A tentative selection of the six most promising systems was based on gross scores against a series of qualifying factors.

Because each application must meet unique conditions for a given house, geographic location and climate, no system can be selected as best for all conditions. In some cases, systems not included in the six tentative selections may prove more desirable and the choice should be left open. A full discussion of the twenty-one systems and methods of analysis is included in Section II.

Conclusions.

The six tentative system selections are:

SYSTEM NO. 3 - Air collector, rock storage, unitary or split heat pump of any generic type, solar energy used for direct heating only.

SYSTEM No. 6 - Identical to System No. 3 but employing a liquid collector and water storage.

SYSTEM No. 19 - Centralized liquid collector and water storage, incremental water-to-air heat pumps connected to a common hydronic loop.

SYSTEM NO. 17 - Liquid collector, water storage, unitary water-to-air heat pump.

SYSTEM No. 1 - Air collector, rock storage, unitary air-to-air heat pump.

SYSTEM No. 4 - Identical to System No. 1 but employing a split air-to-air heat pump.

1.3. PHASE II

Summary.

Systems 3 and 6 have been designed in greater detail and studied in relation to a typical single family dwelling in Little Rock, Arkansas. Load profiles have been developed and five sizes of each system have been matched against them to determine yearly energy use and solar participation. Costs for each size and type of system have been estimated and, together with energy costs, have been used to determine cash flow and cost effectiveness.

System 3 and System 6 were compared for equal collector area and equal solar participation. In each case System 6 had a lower first cost and superior financial benefit.

Conclusions.

System 3, employing air collectors bears the penalties of higher collector cost, higher retrofit costs and lower solar participation for a given size system.

The conclusions of this report, therefore, are that System 6, employing liquid collectors, should be selected for retrofitting USAF houses in Little Rock, Arkansas.

It should be recognized, however, that the installation cost penalties of System 3 will not occur in new house construction, and that this system may well be viable under these circumstances.

The costs of air collectors used in this report are those of the only two major manufacturers (Solaron and Sunworks). Subsequent to compiling this report another manufacturer (Kalwall) has announced production of a new air collector at two thirds of the current price. Information on this collector is not yet available, but if efficiencies are comparable with Solaron and Sunworks, then the rating of systems 3 and 6 may change. In such a young industry, it is inevitable that advances by manufacturers will change the parameters of the "best" system.

1.4. PHASE IIA

Summary.

Three potential manufacturers of a hybrid heat pump system have been identified and the performing curves of one manufacturer have been used in compiling this report.

A solar-assisted hybrid heat pump heating and cooling system has been designed and its performance in various modes of operation has been studied in relation to a typical single family house both in Little Rock, Arkansas, and Scott AFB, Illinois.

Load profiles for each house have been developed and various sizes of solar systems, integrated with one heat pump unit, have been matched against them to determine energy consumption. Costs for each system have been estimated and, together with energy costs, have been used to determine cash flow and cost effectiveness.

Conclusions.

At Little Rock AFB, the solar-assisted hybrid heat pump system (System #22) consisting of a combination of liquid solar collectors integrated with a heat pump capable of operating in either the water-to-air or air-to-air modes consumes approximately 5 percent more energy annually than the solar-augmented heat pump system (System #6). At Scott AFB, the hybrid heat pump system consumes approximately 30 percent more energy in January than the solar-augmented heat pump system. These results stem from the fact that the hybrid system operates for substantially fewer hours in the direct solar heating mode which consumes far less energy than either of the heat pump modes. In addition, there are no indications that the economics of the hybrid system are enhanced by decreasing collector area or by varying storage volume.

1.5. PHASE III

Summary.

A preliminary design integrating System 6 into a specific structure at Little Rock AFB has been completed. Energy conservation measures have been evaluated and a final load profile has been developed. The heat output of the collector array, at various tilts and areas was determined. Costs to purchase and install the system were estimated and the economics of various alternatives were compared to optimize collector area. Preliminary equipment specifications and a recommended equipment list have been developed.

Conclusion.

The preliminary design integrates System 6 into the residence at 170 Alabama, Little Rock AFB. The solar system consists of approximately 145 square feet of semi-concentrating non-evacuated, tubular collector mounted flat on the existing roof. The tubes are rotated to provide an effective tilt of 32°. The major system components are an air-to-air split system heat pump, 330 gallons of thermal storage, an 80-gallon domestic hot water preheat tank, pumps, and controls. The solar system will meet approximately 70 percent of the annual heating and domestic hot water load. Net annual savings at the current electric rate of $0.025/KWH will be about $115.

Recommendations.

It is recommended that System 6 be installed at Little Rock AFB and that it be instrumented to determine system

performance. Additional studies should be conducted to determine applicability of Solar-assisted Heat Pumps/Solar-augmented Heat Pumps Systems at other geographical locations. Also, new developments in heat pump and solar systems should be monitored and their applicability to Air Force needs determined. Developments in Rankine Cycle Heat Pump Systems should be monitored closely. The applicability of multiple heat pumps on one solar system should also be investigated.

SECTION II

PHASE I

2.1. GENERAL

This study concerns the application of solar energy to assist and improve the efficiency of operation of heat pumps in United States Air Force housing units. At the present stage of heat pump technology this implies that solar assist will be used in the heating mode only, and that the cooling load will be met directly using conventional electric drive. There are, however, pilot systems being developed by various manufacturers which will allow solar participation for both heating and cooling modes of operation. These systems, although not immediately available, have been considered and are further described in this report.

Solar energy assist to heat pump operation has been considered either for direct use when the full heating load is met by solar energy, and/or as a low grade temperature heat source to the heat pump in extreme weather, to improve its COP.[1]

2.2 OBJECTIVES

a. The objectives of adding a solar collection system to a heat pump system are:

- To reduce the fossil fuel energy needed to meet the heating load and provide domestic hot water.
- To improve the COP of the heat pump during extreme weather, thus reducing the yearly requirement for auxiliary energy.
- To reduce peak loads by using thermal storage to carry over during periods of high demand.
- To reduce peak heating and cooling loads, and yearly energy use, improving the thermal qualities of houses.

---

1. COP - Coefficient of performance - defined as heat output (heat delivered to house) divided by energy input to operate heat pump, expressed in the same units.

. To reduce the operating and life cycle costs of the heat pump system.

. To use solar energy as an alternative to non-renewable energy resources.

b. The objectives of this study are:

. To develop a set of generic solar assisted heat pump systems from which selections can be made for different sizes and types of houses in various geographic locations with widely differing climates.

. To select a system or systems and prepare designs for installation in an existing house or houses at Little Rock Air Force Base.

. Although not included in the original scope of work, a further objective of this study should be to install a pilot system that incorporates the latest technological developments. This installation would be experimental, but would foster improvements in the domestic application of both heat pumps and solar energy.

2.3. METHODOLOGY

The methodology used in compiling this report was to assemble information from manufacturers, from existing heat pump systems, and from the contractor's own extensive technical labrary, and use this information as the basis for developing potential systems and assessing their merits.

All judgments made in this report are subjective and are based on the experience of the contractor and on information from the above sources.

Manufacturers of heat pumps were contacted for data on their latest models. Their responses revealed that the new generation of heat pumps generally have higher COPs and are able to operate over a larger range of outdoor temperatures than models manufactured two or three years ago. Manufacturers are also becoming conscious of EER (Energy Efficiency Ratio) ratings, and values of 7-8 for cooling are common. The performance of the Westinghouse HI-RE-LI heat pump is typical and has been used when making subjective judgments of generic machines.

Manufacturers of solar collectors were contacted for data on construction and performance. The performance data

were reduced to the same base and the efficiency curves shown on Figure 1 were developed. These curves compare the relative performance of collectors, and help reflect the most favorable collector to be selected for a given set of operating and climatic conditions.

The contractor was retained by HUD to assist in managing its solar energy program, and part of his task was to review actual designs and installations of solar energy systems in houses located in various parts of the country. Of the 50 designs reviewed, nine were solar assisted heat pump systems. Experience gained in this work was applied in compiling this report.

Little Rock AFB was visited and base personnel were asked about their experiences with the existing heat pump systems. The systems on this base have been in continuous use for over 15 years, and the benefit of experiences gained has been applied in compiling this report.

The feasibility of using low temperature turbine Rankine cycle heat pumps was investigated.

Systems using air as a refrigerant were investigated to determine their feasibility and date of availability.

The use of Stirling engines as heat pumps was also investigated.

Over the past year the contractor has worked in conjunction with others to determine the causes of corrosion in solar energy systems, particularly those with aluminum absorber plates and has cooperated with them to develop a water treatment method for corrosion control. Results of this treatment at the George A. Towns School in Atlanta indicate that corrosion can be limited to less than 0.1 mil per year. This cooperative effort is an ongoing task and the experience gained was applied in these systems evaluations.

Various universities and research organizations were contacted to determine:

- The progress in developing a suitable phase change storage material.

- Whether phase change materials are presently available for commercial use.

- Whether the characteristics of phase change materials will complement the operation of a solar assisted heat pump system.

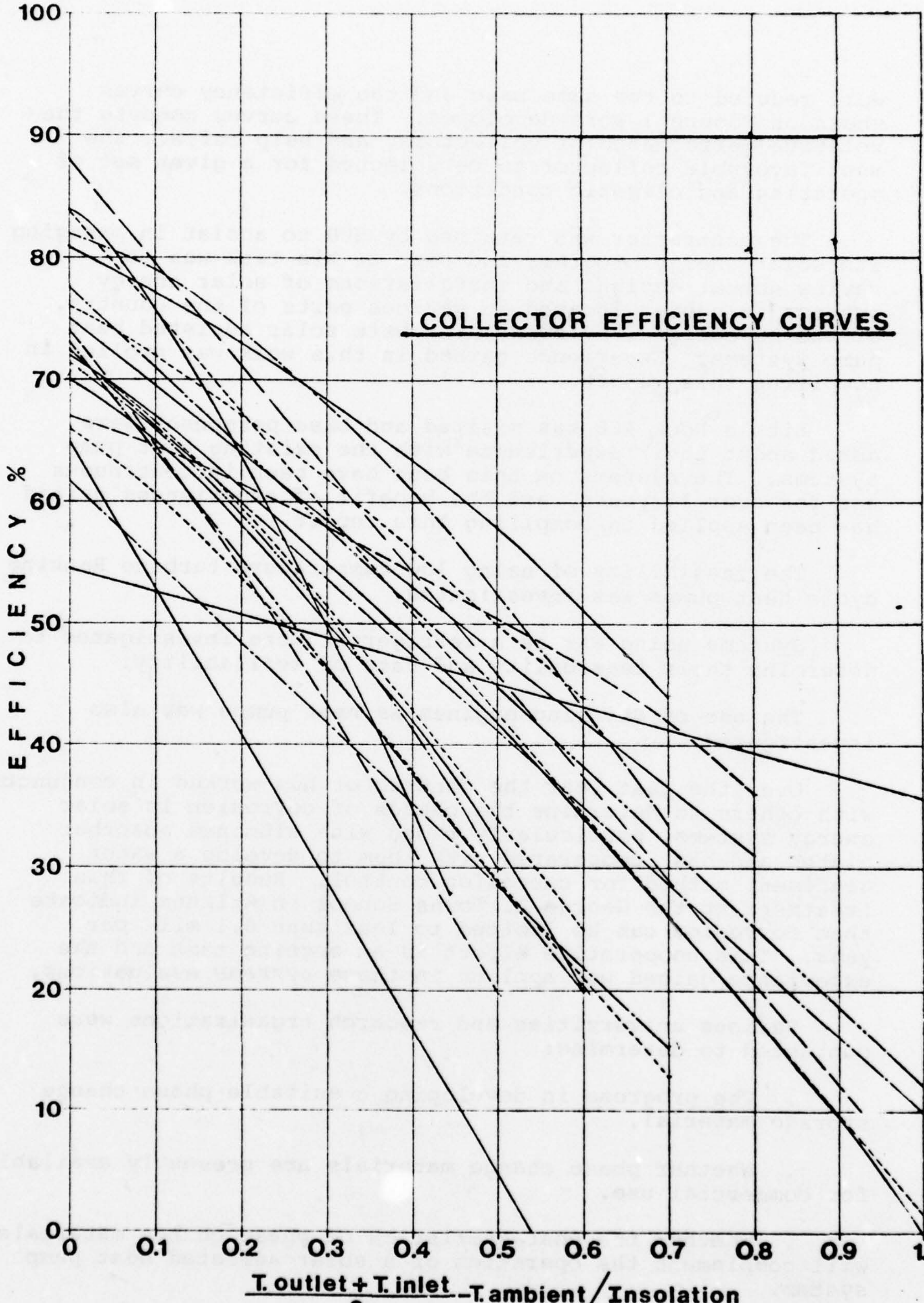


Figure 1. Typical Collector Efficiency Curves

2.4. COMMON DESIGN CONSIDERATIONS

There are design considerations common to various systems which, while not directly affecting the operation of the heat pump, are important factors and must be considered.

a. Frost Protection

All liquid collector systems are susceptible to frost damage. The two most common methods of protection are to use an antifreeze/water mix or a drain-down system. The use of antifreeze fluids introduces complicating factors of corrosion control and separation of the antifreeze circuit from the rest of the system. This separation implies the use of heat exchangers which require an approach temperature differential to make them work. This temperature differential either reduces the usable temperature of collected heat or forces the collector to operate at a higher temperature and consequent lower efficiency. Present federal policy requires the safety feature of double heat exchangers to separate antifreeze circuits from potable water supplies with the consequent double loss effect of approach temperature differential.

Frost protection by drain-down is achieved by maintaining a free air space in the storage tank and pumping water into the collectors and back to the storage tank whenever favorable collection conditions exist. When heat collection is no longer possible, the pump stops. Automatic air vents at the highest point of the system open and allow all the water within the collector system to drain back into the storage tank. This method of frost protection is more direct and requires less equipment than a system employing antifreeze. It also overcomes the disadvantage of heat exchanger approach temperatures. The system must, however, be carefully designed and pipes installed at the correct pitch to insure full and positive draining of the collectors, or ice formation and rupture of the absorber plate will occur. It is recommended that wherever possible, drain-down frost protection be employed on the grounds of simplicity and lower cost. All system diagrams show drain-down frost protection. Details of optional frost protection are shown in Figure 2.

Air collectors are not subject to frost damage. When incorporated into systems that include air-to-water heat exchangers, care should be taken to prevent cold air gravity circulation from the collectors with the attendant danger of freezing the heat exchanger. Careful location of the heat exchanger, proper ductwork routing, and close-fitting dampers will avoid this problem.

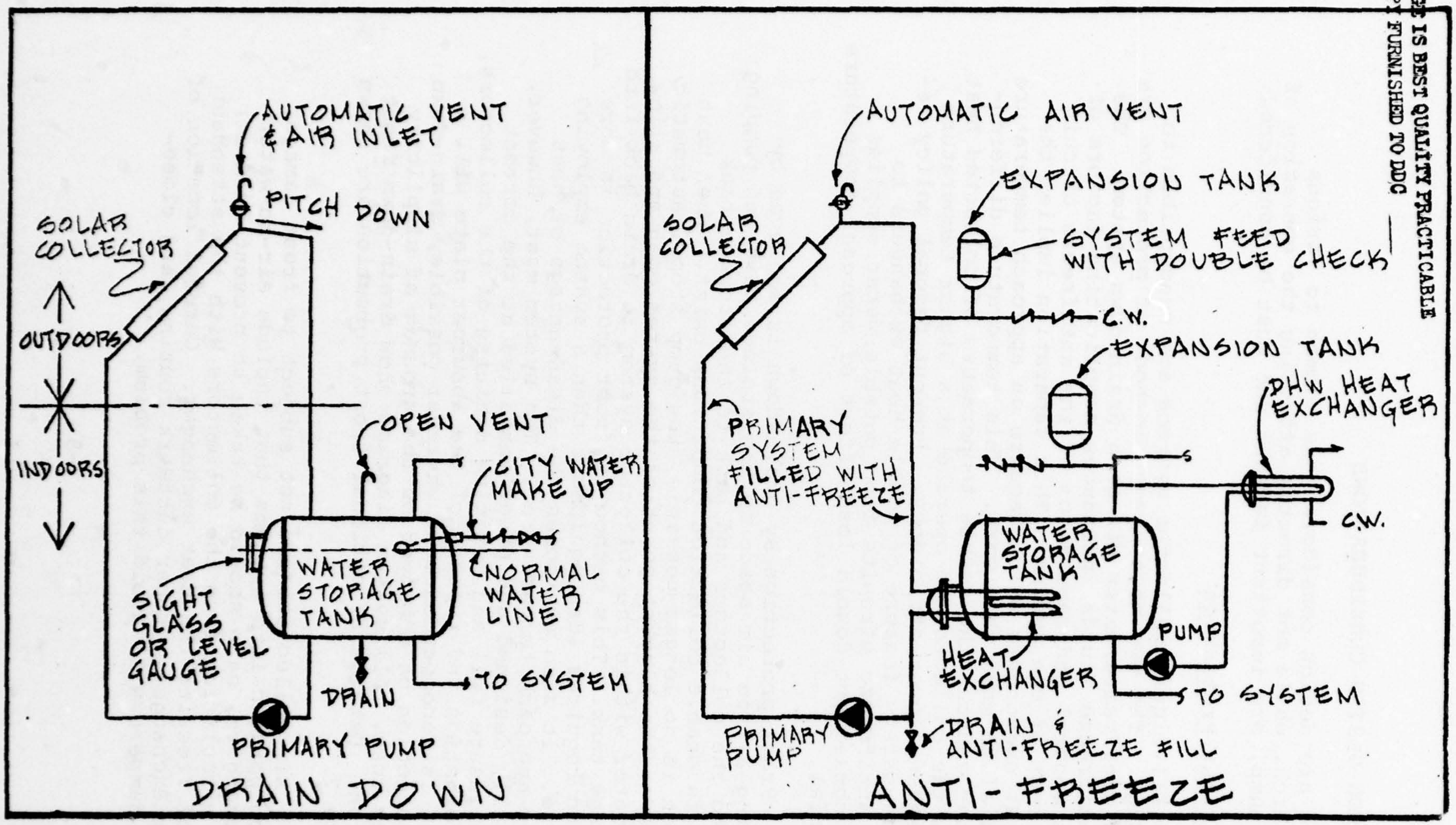


Figure 2. Optional Methods of Frost Protection

b. Storage

All systems considered in this report require some means of thermal storage. The thermal capacity of the storage is theoretically a function of the magnitude of the loads, the system operating temperature range, the area of solar collectors, and the length of time stored heat is intended to meet the load. In a retrofit situation, more practical concerns, such as space available and difficulty of installation, may dictate the selection of a particular type and size of storage. The following three types of storage have been considered when selecting systems:

(1) Phase-changing Materials.

(2) Rocks.

(3) Water.

Each type is discussed, in turn, below:

(1) Phase-changing Materials. There have been no recent breakthroughs in phase change materials for thermal storage. There is continuous effort among researchers to develop materials suitable for heating, cooling, and domestic hot water system storage.

One of the best materials discovered thus far for heating is Calcium Chloride Hexahydrate. It melts at 83°F, considered optimum for heat pump evaporators. Its latent heat is 45.6 cal/gram (8.2 BTU/lb), and the specific heat is 0.35 BTU/lb-°F for the solid phase (at 68°F) and 0.5 BTU/lb-°F for the liquid phase (at 122°F). The complete chemical costs (including a nucleating agent to prevent super-cooling) is estimated at 4-6¢/lb. It melts congruently, has a high water content (6 hydrates), and is readily available--all considered favorable qualities. Experiments are currently being conducted to determine its corrosive properties although this is not anticipated to be a problem. The most promising use for this material seems to be in thermal storage tubes transferring energy to an airstream directly flowing over the tubes, or encapsulated within some solid material (probably plastic) as building modules (e.g., to form a wall).

So far, good salts suitable for cold storage are very expensive (40¢-60¢/lb.).

Paraffins are considered best for domestic hot water, in spite of higher price (10-15¢lb.), lower latent heat (40 cal/gram = 72 Btu/lb.), and the fact that some abrasion problems have been encountered. Certain acids, such as Naphthalene Benzoic acid eutectic, look promising but research is not sufficiently advanced to draw firm conclusions.

A new development in the storage field stems from recent work done by Dr. Harold Lorsch at the Franklin Institute in Philadelphia under contract with ERDA. To eliminate the problems of low thermal conductivity in phase change materials which necessitates a heat exchanger with a large surface area/volume ratio, he proposes using the heat of precipitation encountered in some chemical mixtures to allow the liquid phase to be maintained. By adding the precipitate to a normal storage tank, the heat capacity can be increased by a factor of 2 to 4 with the same heat exchange characteristics as water and no problems.

Phase change materials and similar approaches to thermal storage are still in the experimental stage, and some have been successfully demonstrated in pilot systems, but are not yet suitable for large-scale or commercial use. For this reason, phase change materials are not considered for any of the systems discussed in this report. However, these materials may well prove effective in providing large thermal capacity in a small volume within the near future.

(2) Rocks. Of the systems selected and discussed in this report, all of those employing air collectors may also employ rock storage as this provides a low cost and simple means of storage. The disadvantage of using rocks, however, is that rocks have a lower specific heat and conductivity than water and require approximately three times the space for the same thermal capacity. Ideally, rock storage should be as close as practicable to the collectors, as air ducts must be used for connections between them. This is, however, not always possible in an existing building, and if there is insufficient space within the house, and the rock storage bin has to be constructed away from the house, it may be extremely difficult and costly to make the necessary ductwork connections.

(3) Water. All of the selected systems using liquid collectors also use water as the thermal storage medium. The water is normally contained within a metal tank, either pressurized or open to atmosphere. The tank can be accommodated in any space available within or around the house, or even underground because only relatively small pipes are required to connect the storage tank to the collectors and the rest of the system.

In summary, only two types of storage are considered in this report: rocks and water. Generally, rock storage, in conjunction with air collectors, is the simplest and cheapest solution for systems which use warm air directly and do not require high temperatures.

Some retrofit situations will, however, preclude the use of rock storage because of space problems and the difficulty of running air ducts. In these cases, it may be more practical and economical to use water storage, even though the overall liquid system has a lower rating. For systems that require high temperatures, such as the Rankine Cycle Heat Pump, water is the only practical storage medium.

2.5. SOLAR COLLECTORS

Three generic types of solar collectors have been considered in this study:

a. Flat Plate Liquid.

b. Flat Plate Air.

c. Tubular.

There are many variations within each group, such as type and number of cover plates, type of absorber, absorber coating, and insulation, that affect performance, cost, and ease of installation.

New types of high efficiency focusing collectors are being developed, but are precluded from this study on the grounds of their high cost, experimental nature, and inability to capture diffuse radiation.

Figure 1 shows typical relative instantaneous efficiencies for currently available collectors.

Collectors should be selected for high efficiencies over normal operating range. For example, collectors used primarily for pre-heating require high efficiency at low collection temperatures, but do not need to collect efficiently at high temperatures; whereas, systems such as the Rankine Cycle Heat Pump require higher operating temperatures and the selected collector should be capable of collecting efficiently under these conditions. Referring to Figure 1, collectors for pre-heating should have high efficiency for an abscissa value between 0.1 and 0.3. Collectors used

predominantly for direct heating should have high efficiency for abscissa values between 0.3 and 0.6. Collectors used for any high temperature application (above 160°F) should have high efficiency for abscissa values above 0.7.

Flat plate collectors are available with one or two cover plates of transparent material, usually glass, but sometimes plastic. The selection of a particular collector construction should be related to climate and end use. Two cover plates are not required for low temperature collection in warm climates and may, in fact, be detrimental as each cover plate reflects a fraction of the available sunshine before it can reach the absorber plate. Two cover plates should be used when the temperature difference between the absorber plate and the outdoor air is high (greater than 80°F) as occurs in severe winter climates for heating, or when collecting at high temperatures for cooling.

Various surface treatments are available for the absorber plate. The least costly and most durable to date is a flat black surface which has a high absorptivity (greater than 0.93) but also has a high emissivity. Selective surfaces are available at a higher cost (approximately 50¢ per square foot extra) which have a slightly lower absorptivity (0.9), but a very much lower emissivity (0.2). This lower emissivity reduces radiation losses which are a function of plate temperature. Radiation losses, however, only become significant at relatively high absorber plate temperatures (over 160°F). For low temperature applications, flat black surfaces are acceptable, less costly, and sometimes more desirable because of their higher absorptivity.

The type and slope of roof will affect the collector installation. Ideally, flat plate collectors should be installed at a tilt of latitude +10° for heating and -10° for cooling. Small angular variations in tilt will not significantly affect yearly collector output, but some collectors are more sensitive to this than others. If the roof slope angle is shallow, it may be necessary to build a superstructure or change the roof pitch to achieve an acceptable collector tilt angle. KTA makes a high performance liquid horizontal tubular collector which can be mounted at any angle between horizontal and vertical without detracting from its performance. This collector would be eminently suitable for direct mounting on shallow pitch roofs. Air collectors are generally lighter in weight than liquid collectors and are not subject to frost damage. They do, however, require larger roof penetrations for the air duct connections and are usually more difficult to successfully integrate with the existing structure.

Liquid collectors are easier to install and pipe, requiring less alteration to the building, but are susceptible to frost damage.

2.6. SYSTEM TYPE DEVELOPMENT

Heat pump systems fall naturally into four categroies of:

a. Air-to-Air.

b. Water-to-Air.

c. Water-to-Water.

d. Air-to-Water.

Although it is theoretically possible to design many types of systems in each category, practical limitations are imposed by what is or will shortly become commercially available. The bulk of the systems fall into the air-to-air and water-to-air categories with only one water-to-water system currently available.

There are many variations within each generic type and the systems selected as candidates for evaluation were developed by combining major items of equipment to form complete systems; e.g., each heat pump type was considered as a unitary and a split system operating in conjunction with air collectors with rock storage or water collectors with water storage, and with various types of auxiliary energy back-up. Further variations within these selected systems are possible by considering various alternative fuels as back-up energy sources. This determination would be based on the availability of fuels in the particular geographic location under consideration.

Some special discrete systems such as the Rankine Turbine, Stirling Engine, and Rovac, are included in the selection.

The 21 systems developed and selected for evaluation are shown on Table 1, and are illustrated on Figures 3 to 23.

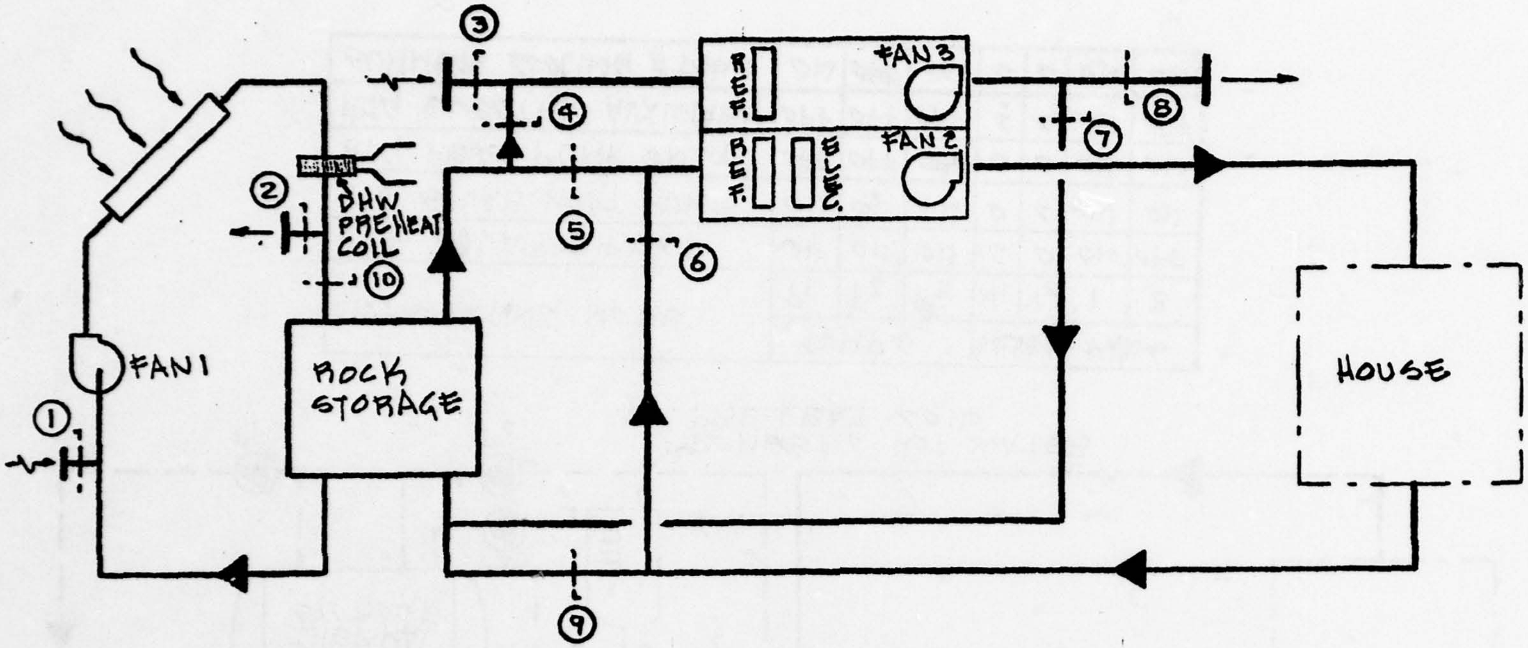
2.7. SYSTEM EVALUATION

To analyze the relative advantages and disadvantages of the 21 systems selected for evaluation, various common qualifying factors were considered. Some of these factors are more important than others, and each one has been weighted

TABLE 1. SYSTEMS EVALUATED

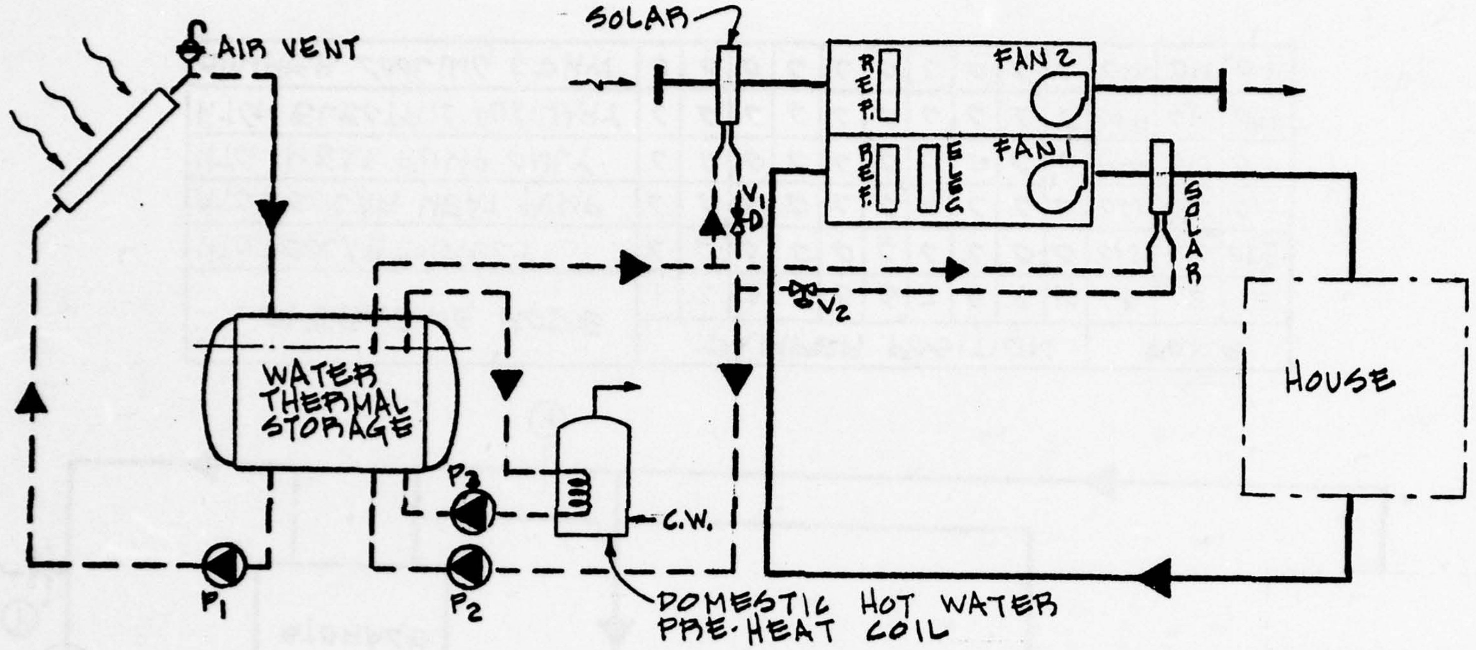
No.	Collector	Storage	Heat Pump	Domestic Hot Water	Auxiliary Back-up	Ref.Fig.	Solar Cooling
1	Air	Rocks	Air/Air Unitary	Air/Water	Electric Coil	4	No
2	Liquid	Water	Air/Air Unitary	Water/Water	Electric Coil	5	No
* 3	Air	Rocks	Air/Air Unitary	Air/Water	Electric Coil	6	No
4	Air	Rocks	Air/Air Split	Air/Water	Electric Coil	7	No
** 5	Liquid	Water	Air/Air Split	Water/Water	Electric Coil	8	No
* 6	Air	Rocks	Air/Air Split	Air/Water	Electric Coil	9	No
7	Attic	Rocks	Air/Air Unitary	Air/Water	Electric Coil	10	No
8	Attic	Rocks	Air/Air Split	Air/Water	Electric Coil	11	No
9	Air	Rocks	Rovac air cycle	Air/Water	Electric Coil	12	No
10	Liquid	Water	Rovac air cycle	Water/Water	Electric Coil	13	No
11	Liquid	Water	Pivoting tip vane	Water/Water	Electric Coil	14	Yes
12	Liquid	Water	Rankine turbine	Water/Water	Gas or oil	15	Yes
13	Liquid	Water	Stirling heat engine	Water/Water	Gas or oil	16	Yes
14	Liquid	Water	Water/Air Unitary	Water/Water	Gas/Oil boiler	17	No
15	Liquid	Water	Water/Air Unitary Water heat source/ sink	Water/Water	Gas/Oil boiler	18	No
16	Liquid	Water	Water/Air Unitary Ground heat source/ sink	Water/Water	Gas/Oil boiler	19	No
17	Liquid	Water	Water/Air Unitary	Water/Water	Electric Coil	20	No
18	Liquid	Water	Water/Air split	Water/Water	Gas/Oil furnace	21	No
19	Liquid	Water	Water/Air incremental Common hydronic loop	Water/Water	Gas/Oil boiler	22	No
20	Liquid	Water/Ice	Water/Air Unitary Seasonal Storage	Water/Water	Electric Coil	23	No
21	Liquid	Water	Water/Water Unitary	Water/Water	Electric Coil	24	No

* DIRECT SOLAR HEATING ONLY
** SIMILAR TO "JARDINE" SYSTEM



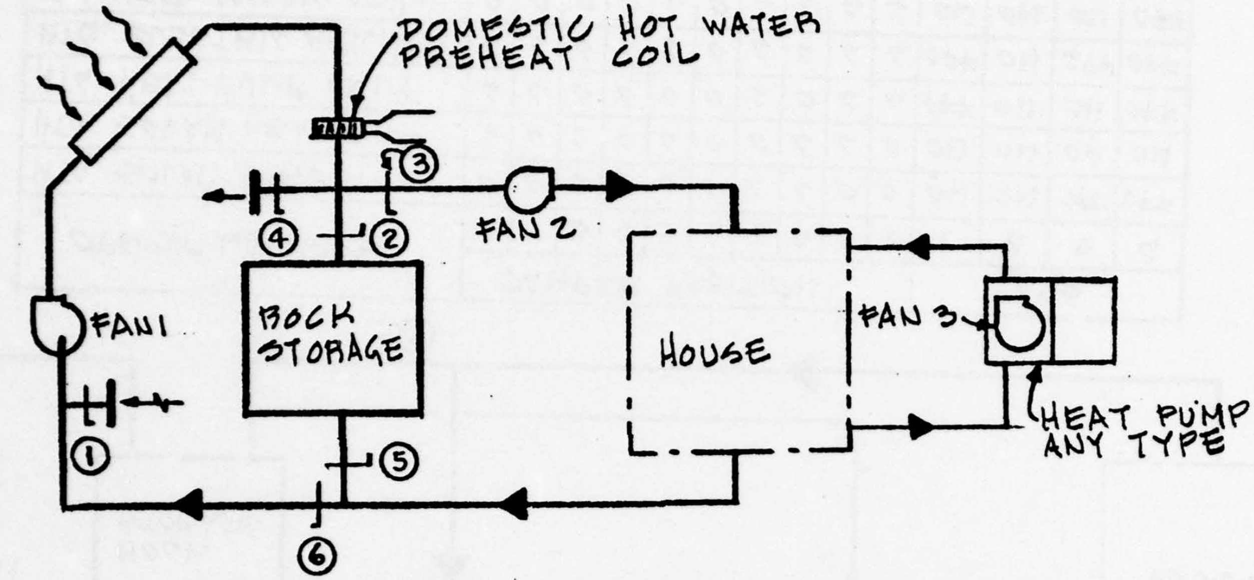
OPERATING MODE	DAMPER POSITION										FANS		
	1	2	3	4	5	6	7	8	9	10	1	2	3
HTG. SOLAR DIRECT	C	C	C	C	O	C	C	C	O	O	ON	ON	OFF
HTG. SOLAR HEAT PUMP	C	C	C	O	C	O	O	C	C	O	ON	ON	ON
HTG. HEAT PUMP ONLY	C	C	O	C	C	O	C	O	C	C	OFF	ON	ON
HTG. ELECTRIC AUXILIARY	C	C	C	C	C	O	C	C	C	C	OFF	ON	OFF
SUMMER COOLING & DHW	O	O	O	C	C	O	C	O	C	C	ON	ON	ON

Figure 3. System No. 1



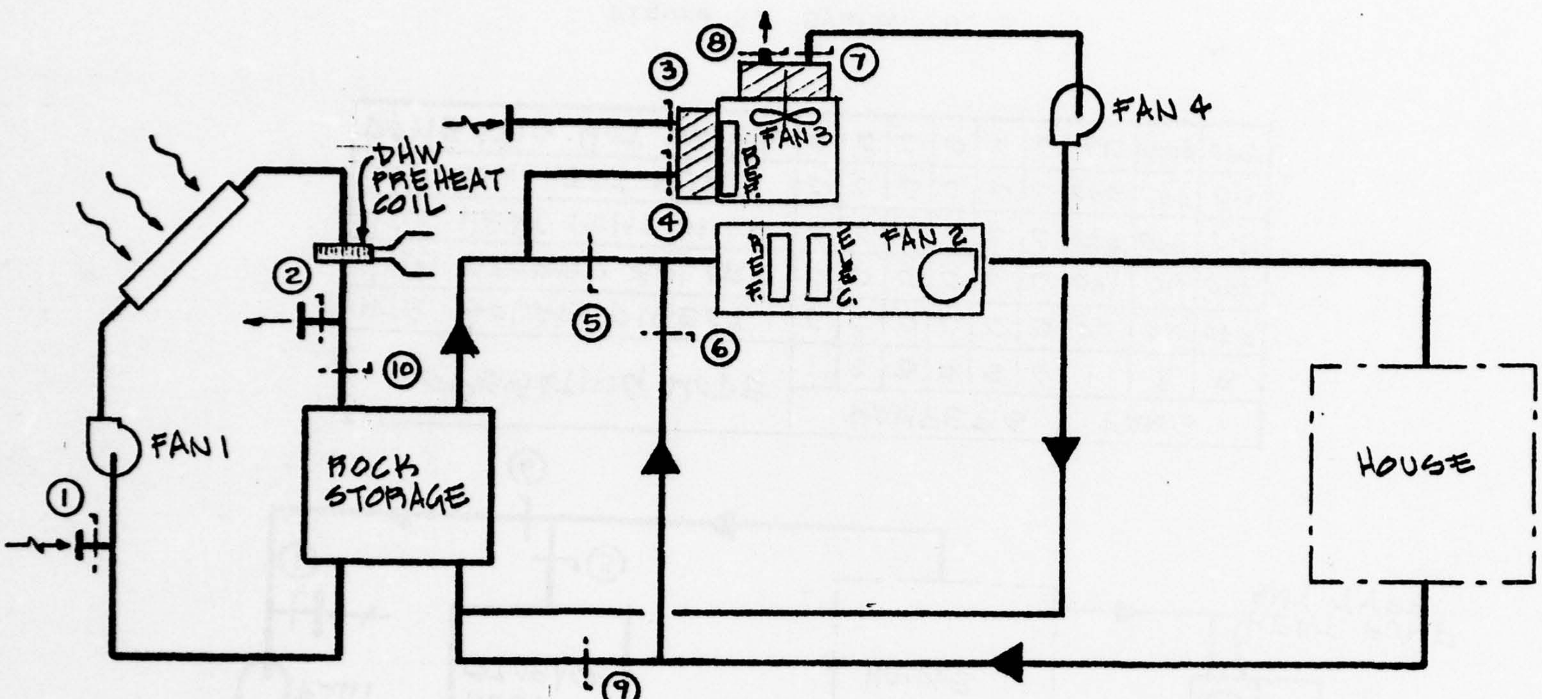
OPERATING MODE	PUMPS			VALVES		FANS	
	P_1	P_2	P_3	V_1	V_2	1	2
HTG. SOLAR DIRECT	ON	ON	ON	C	O	ON	OFF
HTG. SOLAR HEAT PUMP	ON	ON	ON	O	C	ON	ON
HTG. HEAT PUMP ONLY	OFF	OFF	OFF	C	C	ON	ON
HTG. ELECTRIC AUXILIARY	OFF	OFF	OFF	C	C	ON	OFF
SUMMER COOLING & DHW	ON	OFF	ON	C	C	ON	ON

Figure 4. System No. 2



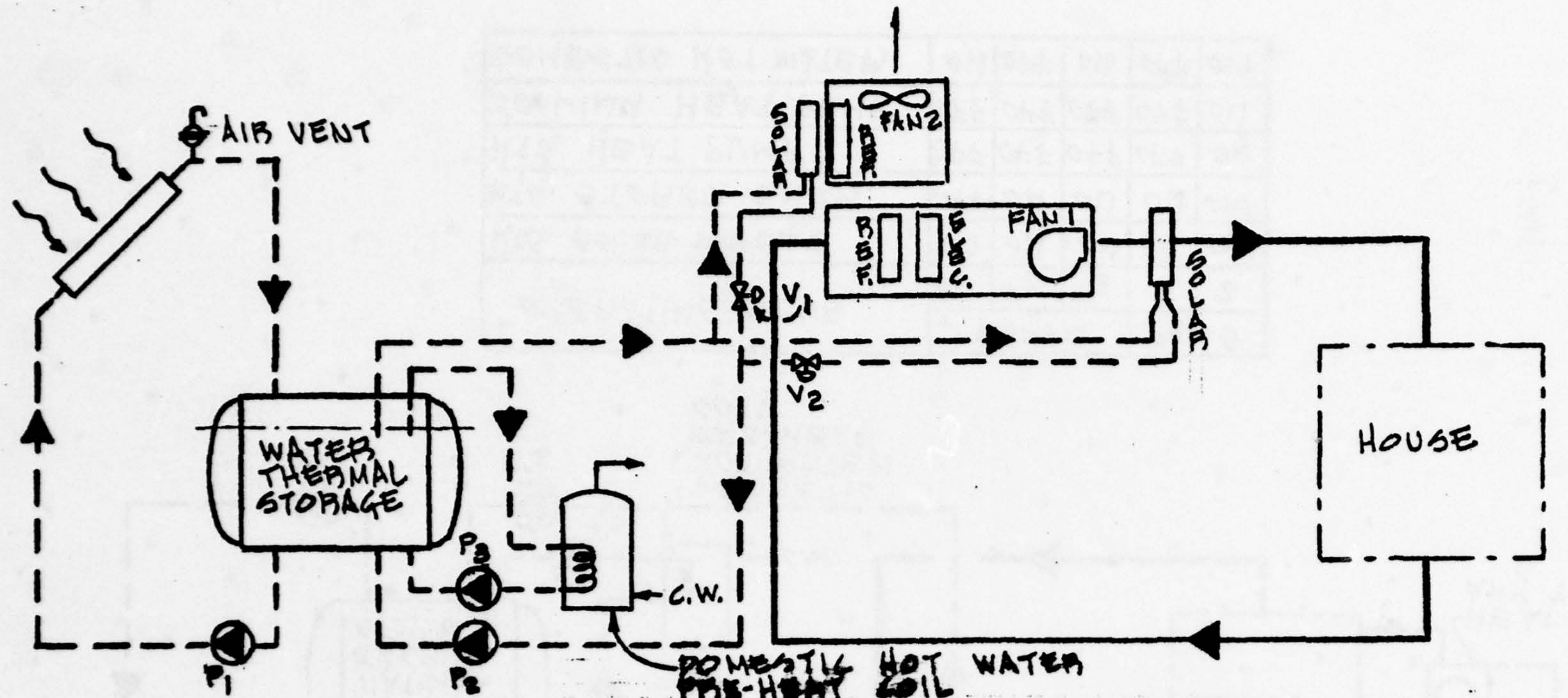
OPERATING MODE	DAMPERS						FANS		
	1	2	3	4	5	6	1	2	3
HTG. SOLAR DIRECT	C	C	O	C	C	O	ON	ON	OFF
HTG. STORED SOLAR	C	O	O	C	O	C	OFF	ON	OFF
HTG. HEAT PUMP ONLY	C	C	C	C	C	C	OFF	OFF	ON
COOLING HEAT PUMP	C	C	C	C	C	C	OFF	OFF	ON
DOMESTIC HOT WATER	O	C	C	O	C	C	ON	OFF	OFF

Figure 5. System No. 3



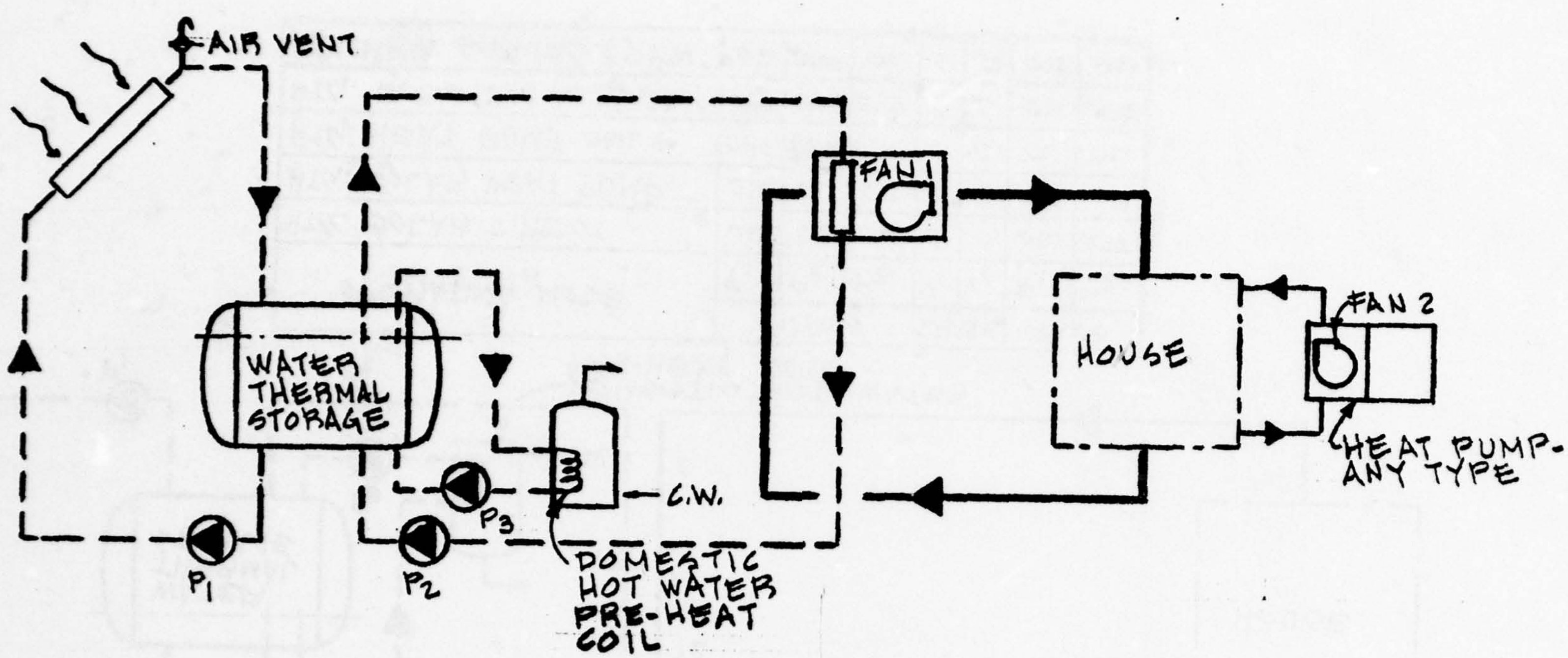
OPERATING MODE	DAMPER POSITION										FANS			
	1	2	3	4	5	6	7	8	9	10	1	2	3	4
HTG. SOLAR DIRECT	C	C	C	C	O	C	C	C	O	O	ON	ON	OFF	OFF
HTG. SOLAR HEAT PUMP	C	C	C	O	C	O	O	C	C	O	ON	ON	ON	ON
HTG. HEAT PUMP ONLY	C	C	O	C	C	O	C	O	C	C	OFF	ON	ON	OFF
HTG. ELECTRIC AUXILIARY	C	C	C	C	C	O	C	C	C	C	OFF	ON	OFF	OFF
SUMMER COOLING & DHW	O	O	O	O	C	O	C	O	C	C	ON	ON	ON	OFF

Figure 6. System No. 4



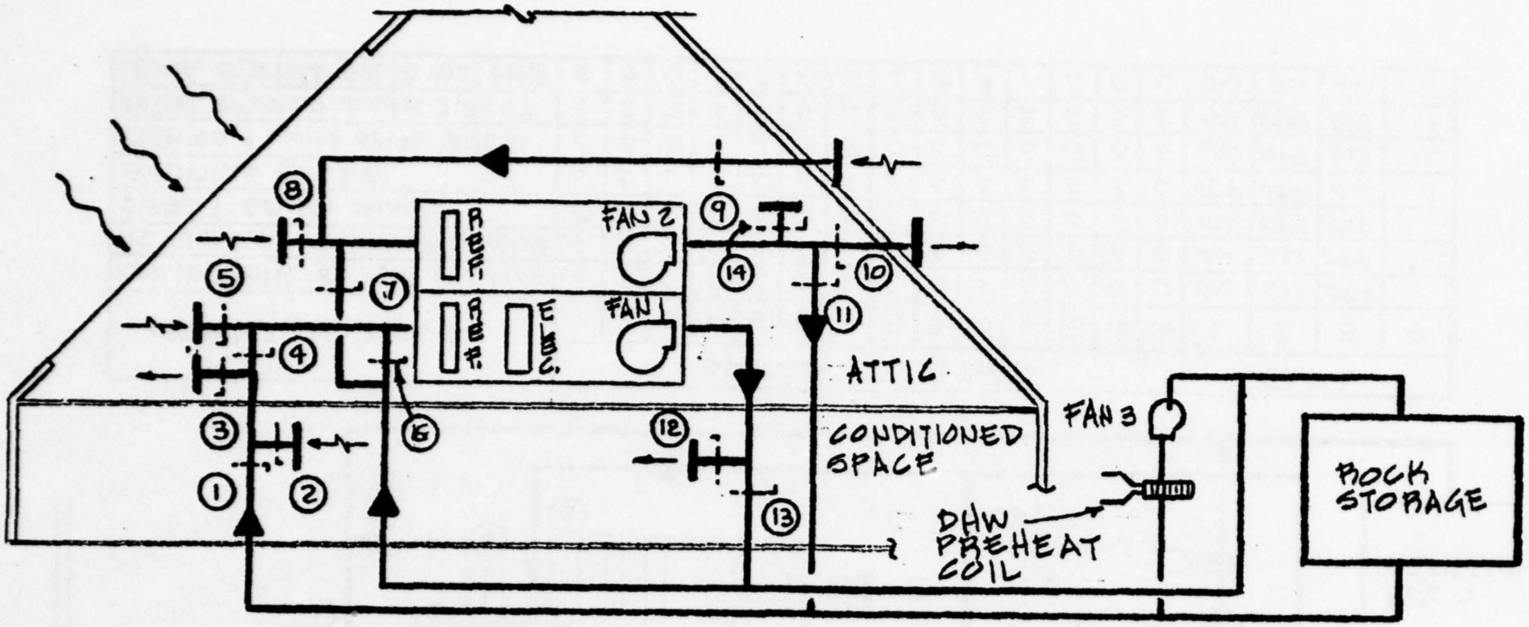
OPERATING MODE	PUMPS			VALVES		FANS	
	P_1	P_2	P_3	V_1	V_2	F_1	F_2
HTG. SOLAR DIRECT	ON	ON	ON	C	O	ON	OFF
HTG. SOLAR HEAT PUMP	ON	ON	ON	O	C	ON	ON
HTG. HEAT PUMP ONLY	OFF	OFF	OFF	C	C	ON	ON
HTG. ELECTRIC AUXILIARY	OFF	OFF	OFF	C	C	ON	OFF
SUMMER COOLING & DHW	ON	OFF	ON	C	C	ON	ON

Figure 7. System No. 5



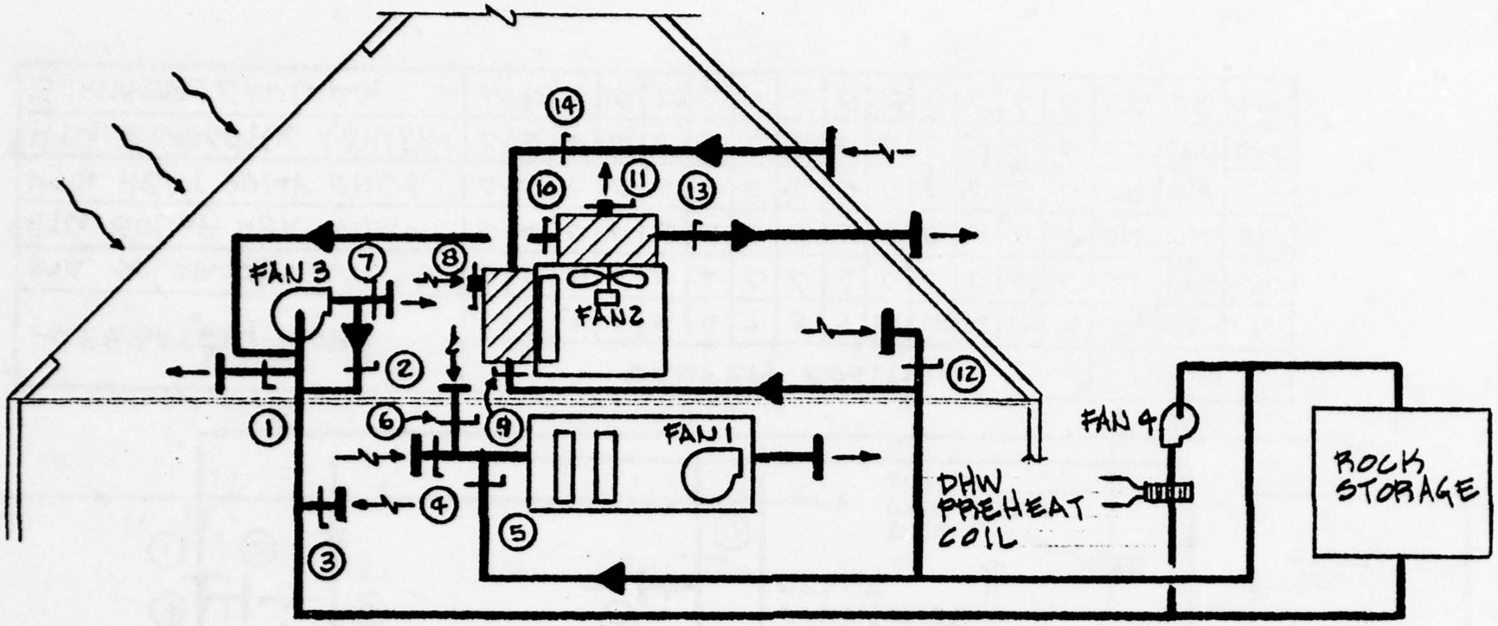
OPERATING MODE	PUMPS			FANS	
	P_1	P_2	P_3	1	2
HTG. SOLAR DIRECT	ON	ON	ON	ON	OFF
HTG. STORED SOLAR	OFF	ON	ON	ON	OFF
HTG. HEAT PUMP	OFF	OFF	OFF	OFF	ON
COOLING HEAT PUMP	OFF	OFF	OFF	OFF	ON
DOMESTIC HOT WATER	ON	OFF	ON	OFF	OFF

Figure 8. System No. 6



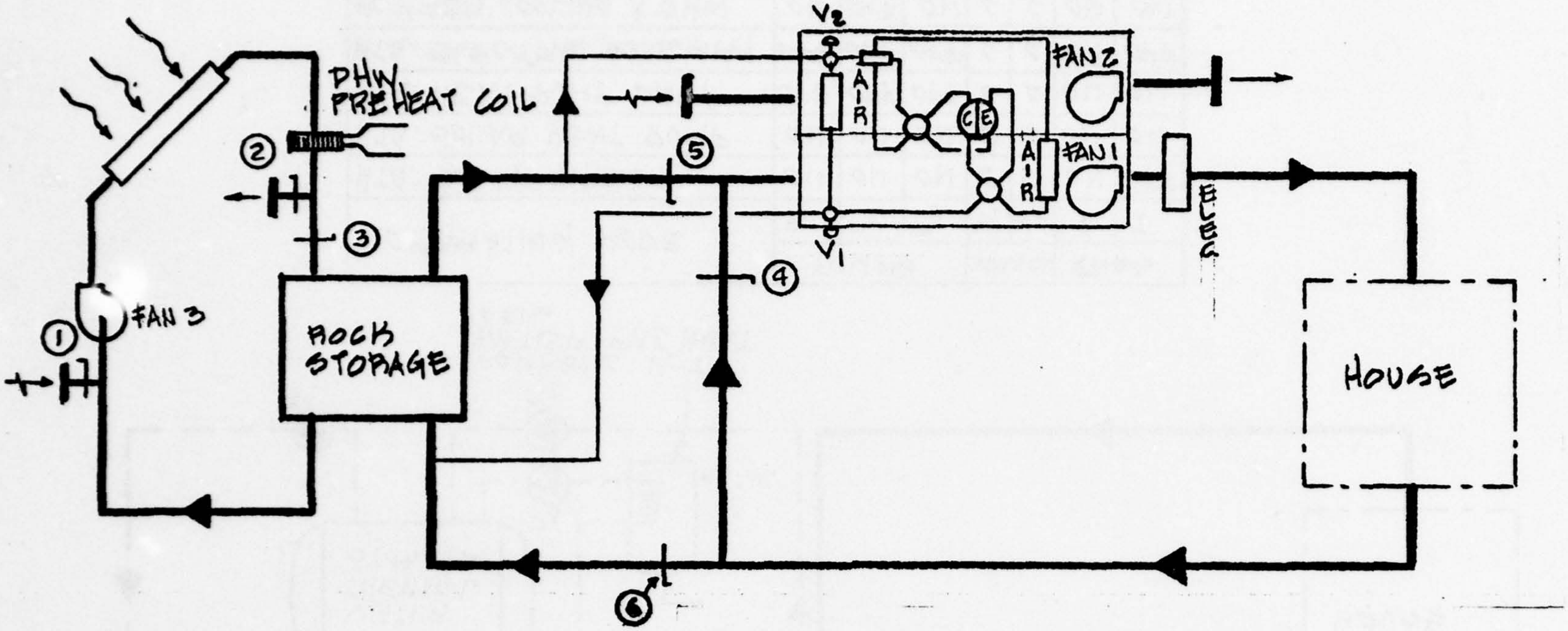
OPERATING MODE	DAMPER POSITION														FANS		
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	1	2	3
HTG. SOLAR DIRECT	C	O	O	C	O	C	C	C	C	C	C	O	C	C	ON	OFF	ON
HTG. SOLAR HEAT PUMP	C	O	O	O	O	C	C	O	C	C	C	O	C	O	ON	ON	ON
HTG. HEAT PUMP ONLY	C	O	C	O	C	C	C	C	O	O	C	O	C	C	ON	ON	OFF
HTG. ELECTRIC AUXILIARY	C	O	C	O	C	C	C	C	C	C	C	O	C	C	ON	OFF	OFF
SUMMER COOLING	C	O	C	O	C	C	C	C	O	O	C	O	C	C	ON	ON	OFF

Figure 9. System No. 7



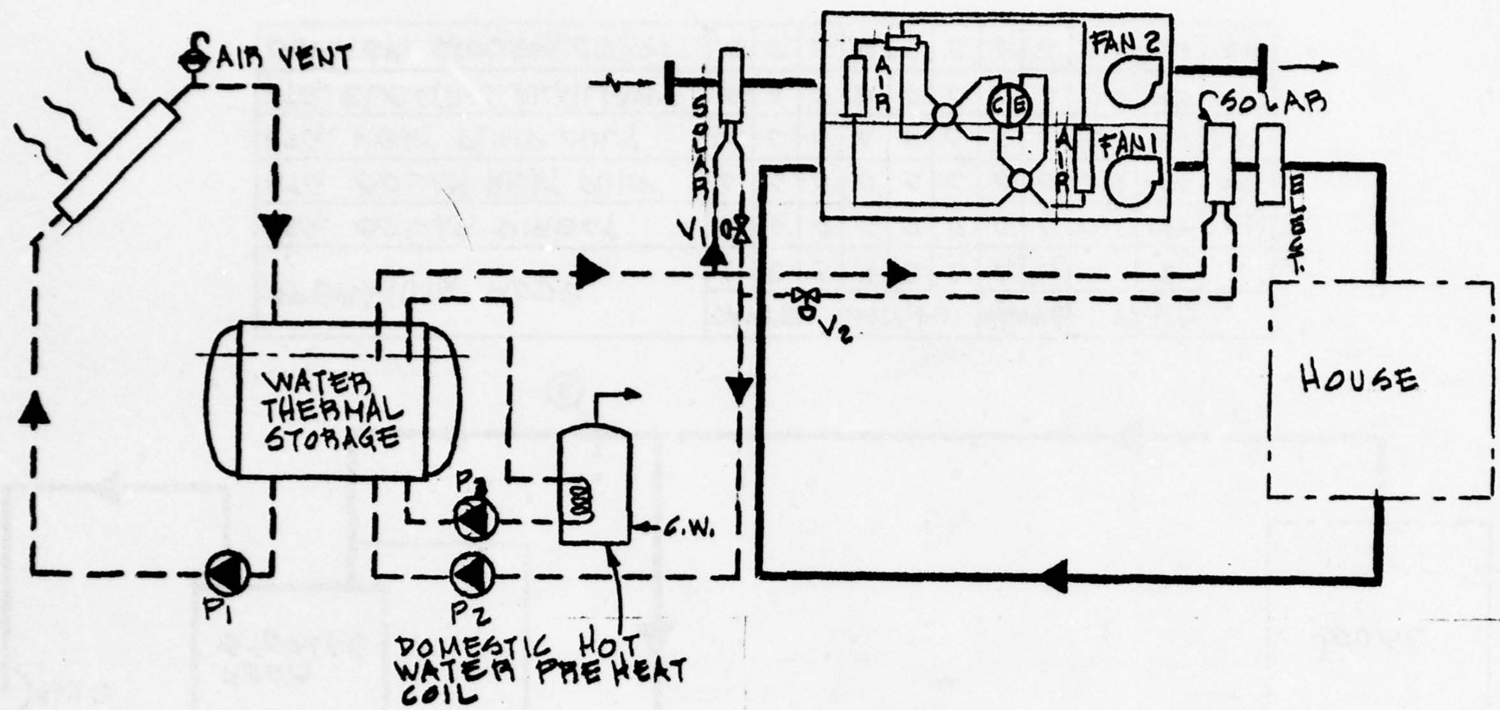
OPERATING MODE	DAMPER POSITION														FANS			
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	1	2	3	4
CONVENTIONAL HT. PUMP (HTG. OR CLG.)	C	C	C	O	C	C	C	C	C	C	C	C	O	O	ON	ON	OFF	
HT. PUMP ATTIC WARMER THAN OUTSIDE	C	C	C	O	C	C	C	O	C	C	O	C	C	C	ON	ON	OFF	
DIRECT SOLAR USAGE	O	C	O	C	C	O	C	C	C	C	C	C	C	C	ON	OFF	OFF	
STORING SOLAR	C	C	C	C	C	C	O	C	C	C	C	O	C	C	OFF	OFF	ON	
STORING & USING SOLAR DIRECT	C	C	O	C	C	O	O	C	C	C	C	O	C	C	ON	OFF	ON	
USING STORED SOLAR DIRECT	C	C	O	C	O	C	C	C	C	C	C	C	C	C	ON	OFF	OFF	
USING STORED SOLAR W/HT. PUMP	C	O	C	C	C	C	C	C	O	O	C	C	C	C	ON	ON	ON	

Figure 10. System No. 8



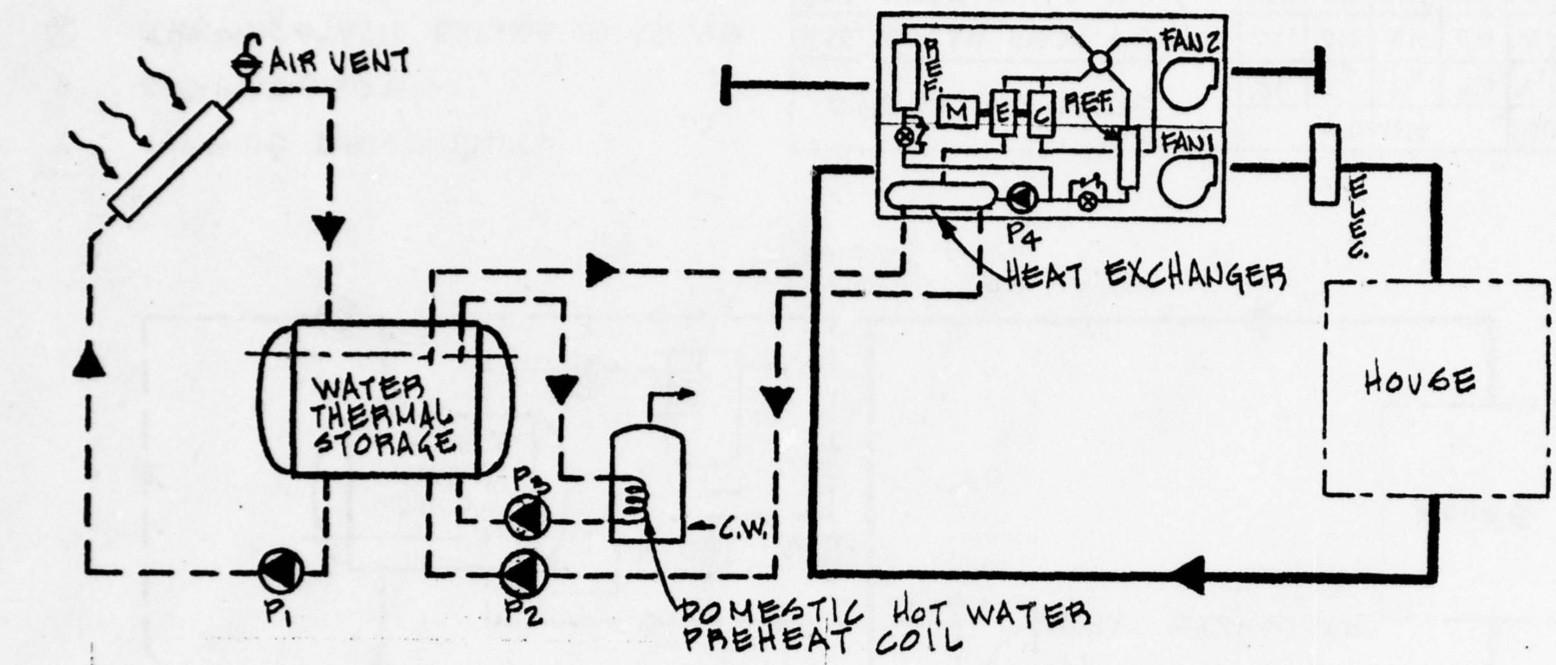
OPERATING MODE	DAMPER POSITION						VALVES		FANS		
	1	2	3	4	5	6	V_1	V_2	1	2	3
HTG. SOLAR DIRECT	C	C	O	C	O	O	C	C	ON	OFF	ON
HTG. SOLAR HEAT PUMP	C	C	C	O	C	C	O	O	ON	OFF	OFF
HTG. HEAT PUMP ONLY	C	C	C	O	C	C	C	C	ON	ON	OFF
HTG. ELECTRIC AUXILIARY	C	C	C	O	C	C	C	C	ON	OFF	OFF
SUMMER COOLING & DHW	O	O	C	O	C	C	C	C	ON	ON	ON

Figure 11. System No. 9



OPERATING MODE	PUMPS			VALVES		FANS	
	P_1	P_2	P_3	V_1	V_2	1	2
HTG. SOLAR DIRECT	ON	ON	ON	C	O	ON	OFF
HTG. SOLAR HEAT PUMP	ON	ON	ON	O	C	ON	ON
HTG. HEAT PUMP ONLY	OFF	OFF	OFF	C	C	ON	ON
HTG. ELECTRIC AUXILIARY	OFF	OFF	OFF	C	C	ON	OFF
SUMMER COOLING & DHW	ON	OFF	ON	C	C	ON	ON

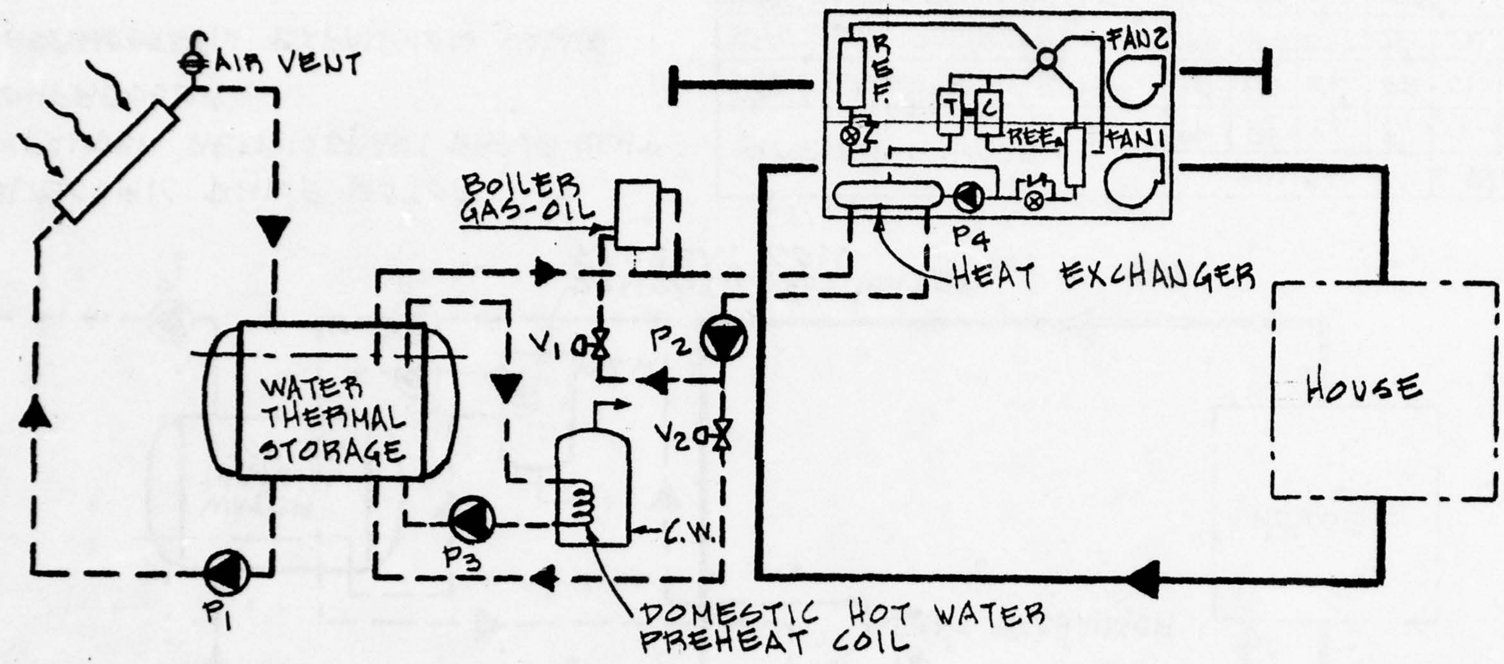
Figure 12. System No. 10



M - ELECTRIC DRIVE MOTOR
E - EXPANDER REFRIGERANT DRIVE UNIT
C - COMPRESSOR
⊗ THERMOSTATIC EXPANSION VALVE

OPERATING MODE	PUMPS				FANS	
	P_1	P_2	P_3	P_4	1	2
HTG. SOLAR HEAT PUMP	ON	ON	ON	ON	ON	ON
HTG. HEAT PUMP ONLY	OFF	OFF	OFF	OFF	ON	ON
HTG. ELECTRIC AUXILIARY	OFF	OFF	OFF	OFF	ON	OFF
SUMMER COOLING & DHW	ON	ON	ON	ON	ON	ON

Figure 13. System No. 11



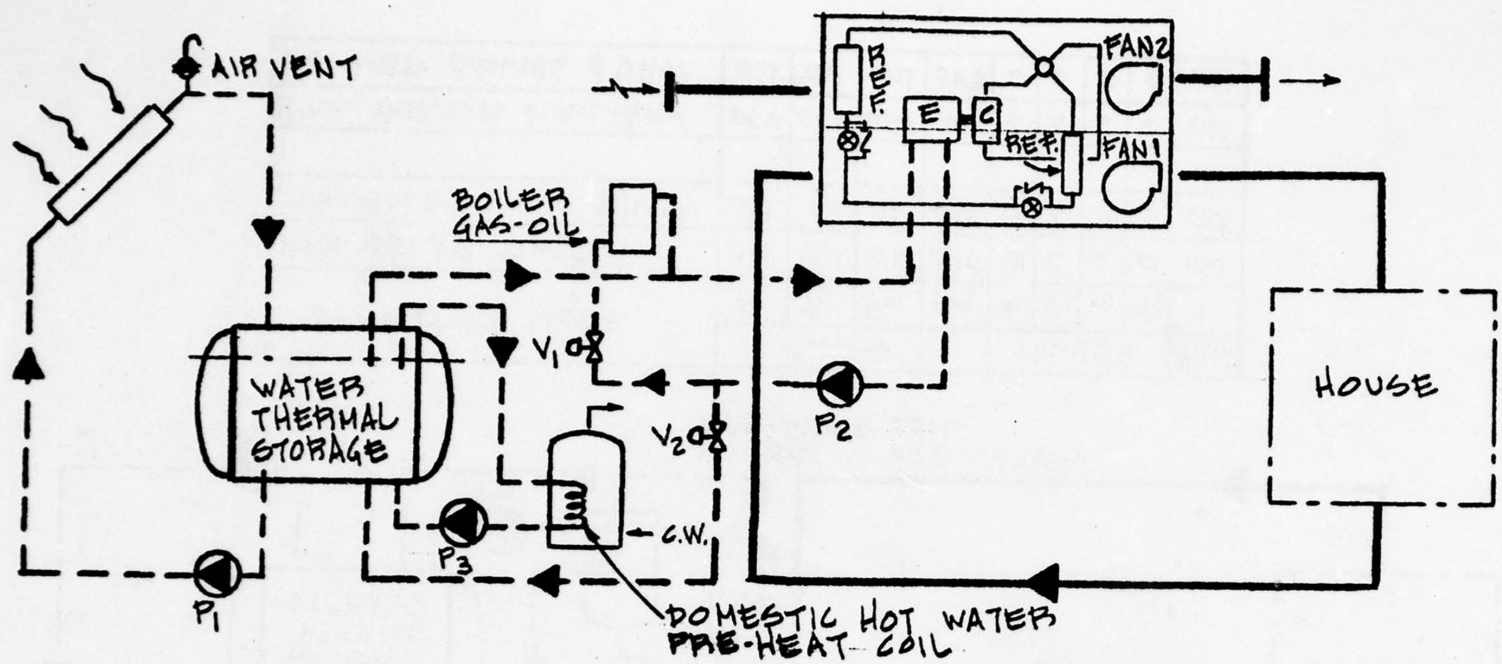
T TURBINE DRIVE UNIT

C COMPRESSOR

⊗ THERMOSTATIC EXPANSION VALVE

OPERATING MODE	PUMPS				VALVES		FANS	
	P_1	P_2	P_3	P_4	V_1	V_2	1	2
HTG. SOLAR HEAT PUMP	ON	ON	ON	ON	O	C	ON	ON
HTG. HEAT PUMP ONLY	OFF	ON	OFF	ON	C	O	ON	ON
SUMMER COOLING & DHW	ON	ON	ON	ON	O	C	ON	ON

Figure 14. System No. 12

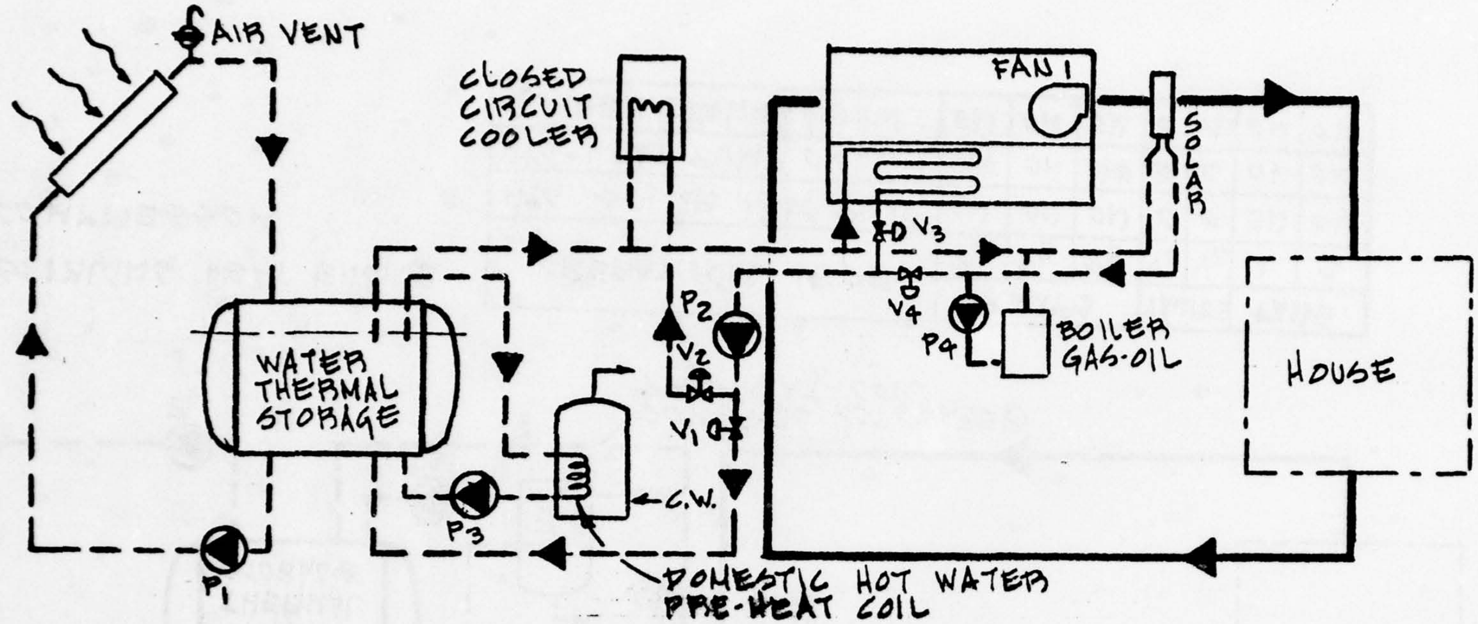


E STIRLING HEAT ENGINE

C COMPRESSOR

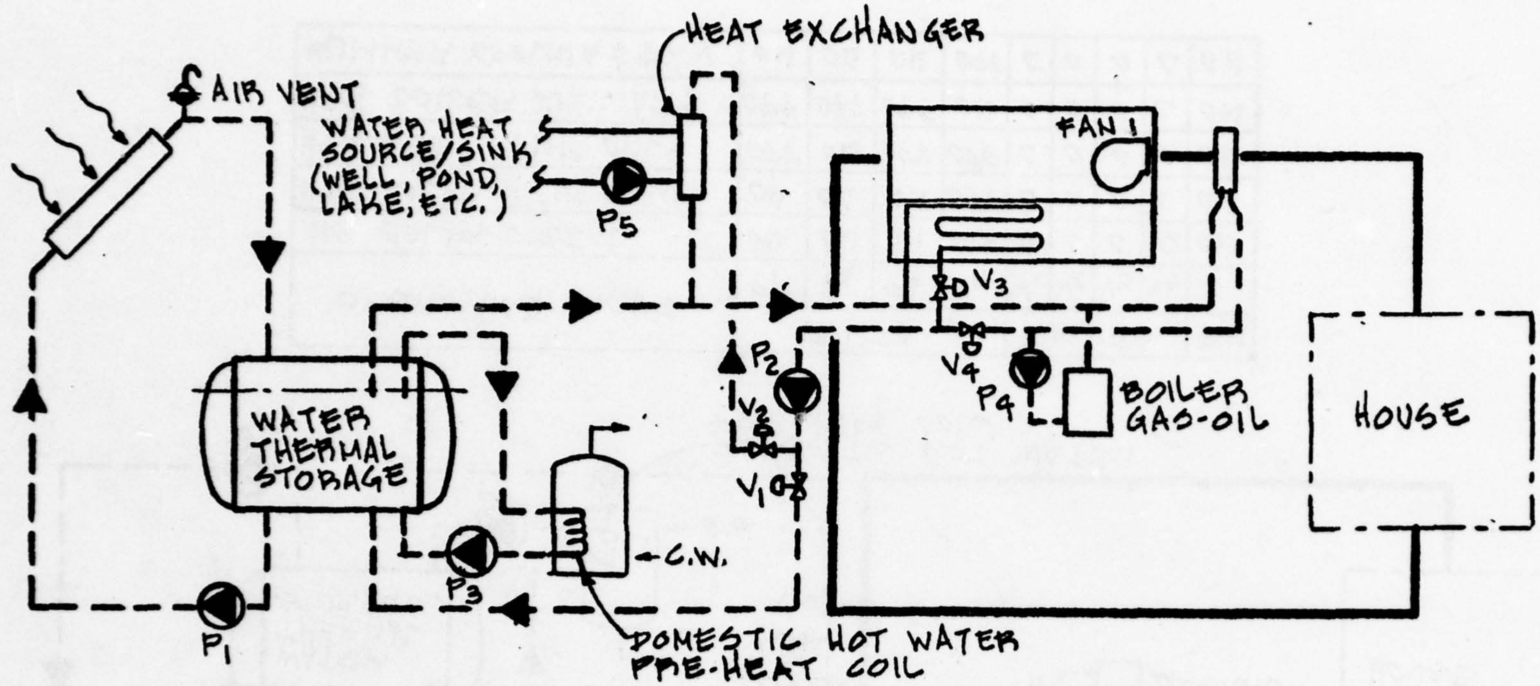
OPERATIONAL MODE	PUMPS			VALVES		FANS	
	P_1	P_2	P_3	V_1	V_2	1	2
HTG. SOLAR HEAT PUMP	ON	ON	ON	C	O	ON	ON
HTG HEAT PUMP ONLY	OFF	ON	OFF	O	C	ON	ON
SUMMER COOLING & DHW	ON	ON	ON	C	O	ON	ON

Figure 15. System No. 13



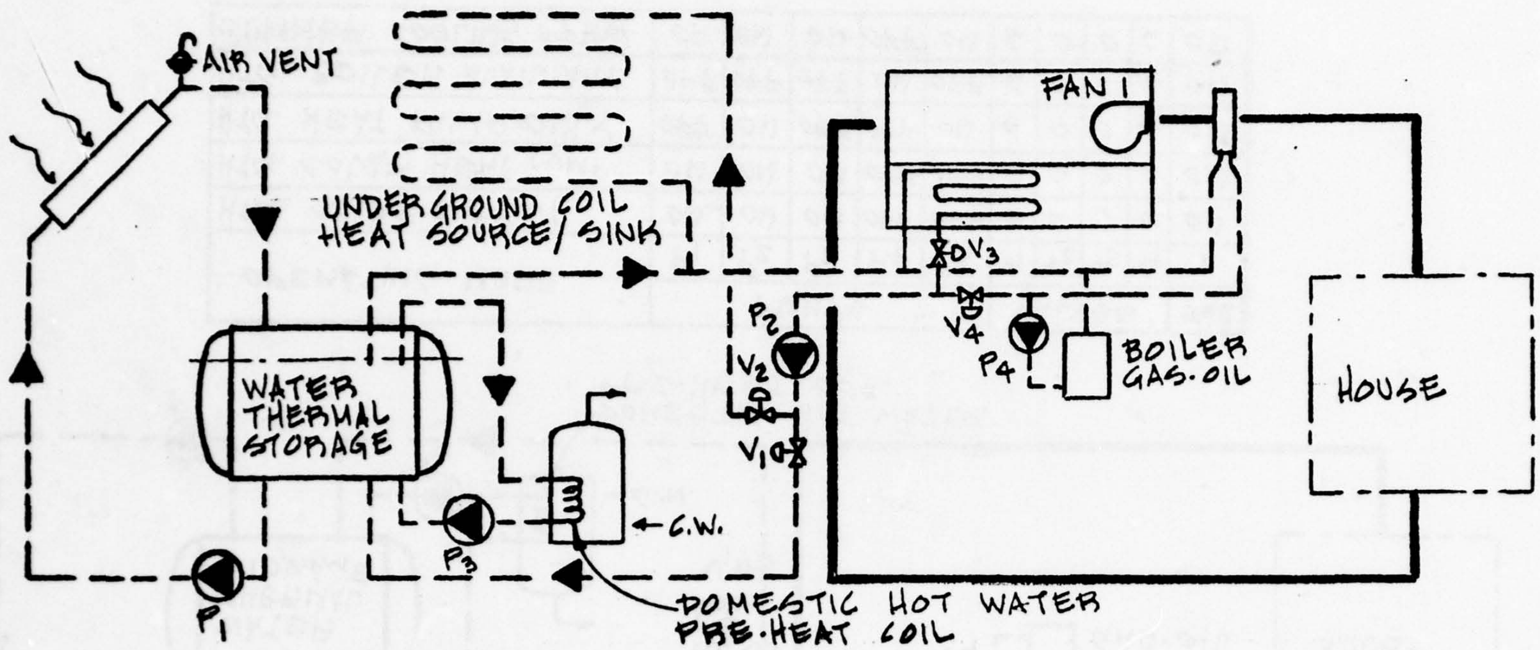
OPERATING MODE	PUMPS				VALVES				FAN
	P_1	P_2	P_3	P_4	V_1	V_2	V_3	V_4	1
HTG. SOLAR DIRECT	ON	ON	ON	OFF	O	C	C	O	ON
HTG. SOLAR HEAT PUMP	ON	ON	ON	OFF	O	C	O	C	ON
HTG. BOILER AUXILIARY	OFF	OFF	OFF	ON	C	C	C	C	ON
SUMMER COOLING & DHW	ON	ON	ON	OFF	C	O	O	C	ON

Figure 16. System No. 14



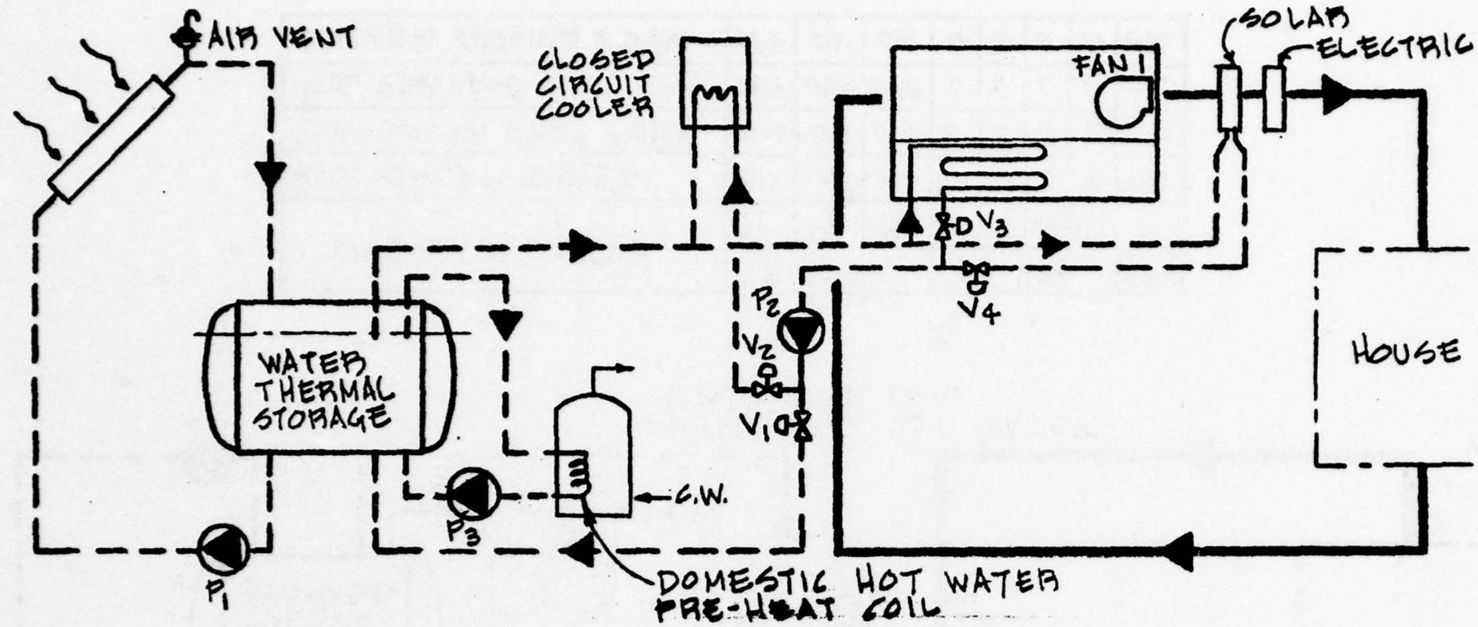
OPERATING MODE	PUMPS					VALVES				FAN
	P_1	P_2	P_3	P_4	P_5	V_1	V_2	V_3	V_4	1
HTG. SOLAR DIRECT	ON	ON	ON	OFF	OFF	O	C	C	O	ON
HTG. SOLAR HEAT PUMP	ON	ON	ON	OFF	OFF	O	C	O	C	ON
HTG. HEAT PUMP ONLY	OFF	ON	OFF	OFF	ON	C	O	O	C	ON
HTG. BOILER AUXILIARY	OFF	OFF	OFF	ON	OFF	C	C	C	C	ON
SUMMER COOLING &DHW	ON	ON	ON	OFF	ON	C	O	O	C	ON

Figure 17. System No. 15



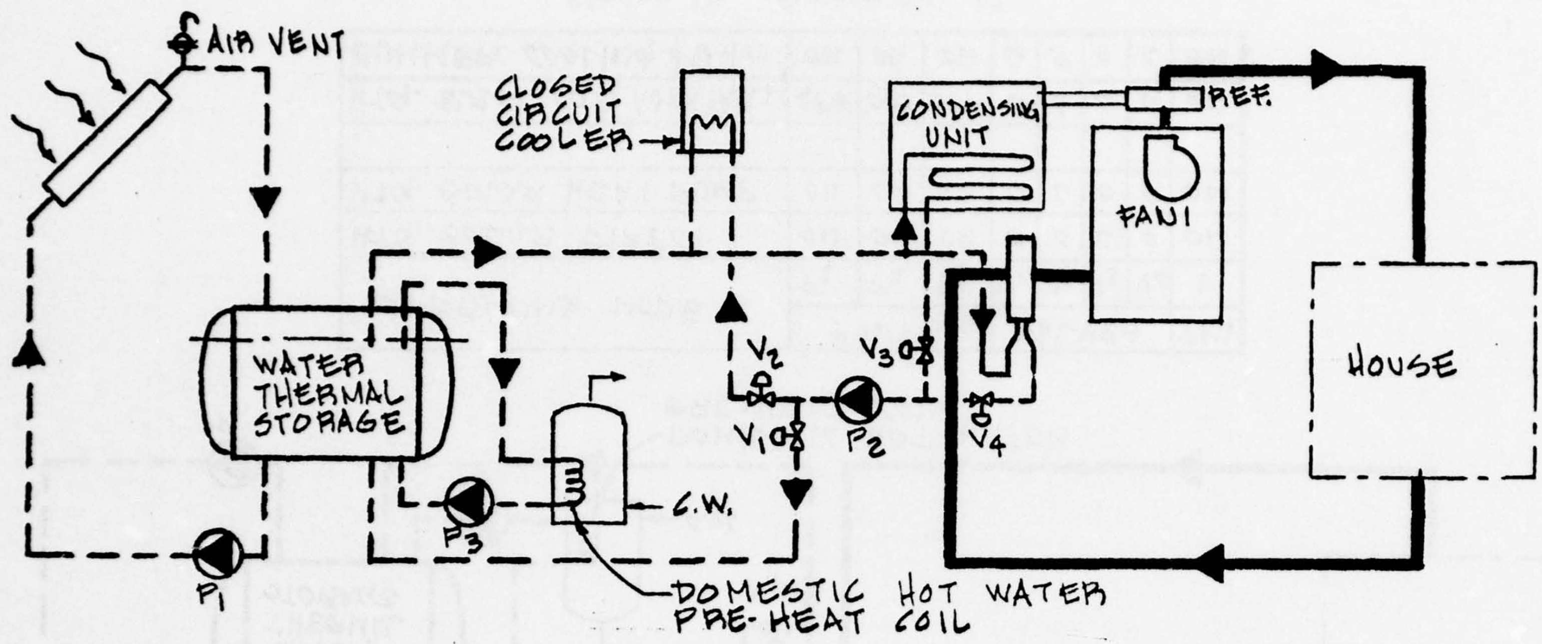
OPERATING MODE	PUMPS				VALVES				FAN
	P_1	P_2	P_3	P_4	V_1	V_2	V_3	V_4	1
HTG. SOLAR DIRECT	ON	ON	ON	OFF	O	C	C	O	ON
HTG. SOLAR HEAT PUMP	ON	ON	ON	OFF	O	C	O	C	ON
HTG. HEAT PUMP ONLY	OFF	ON	OFF	OFF	C	O	O	C	ON
HTG. BOILER AUXILIARY	OFF	OFF	OFF	ON	C	C	C	C	ON
SUMMER COOLING & DHW	ON	ON	ON	OFF	C	O	O	C	ON

Figure 18. System No 16



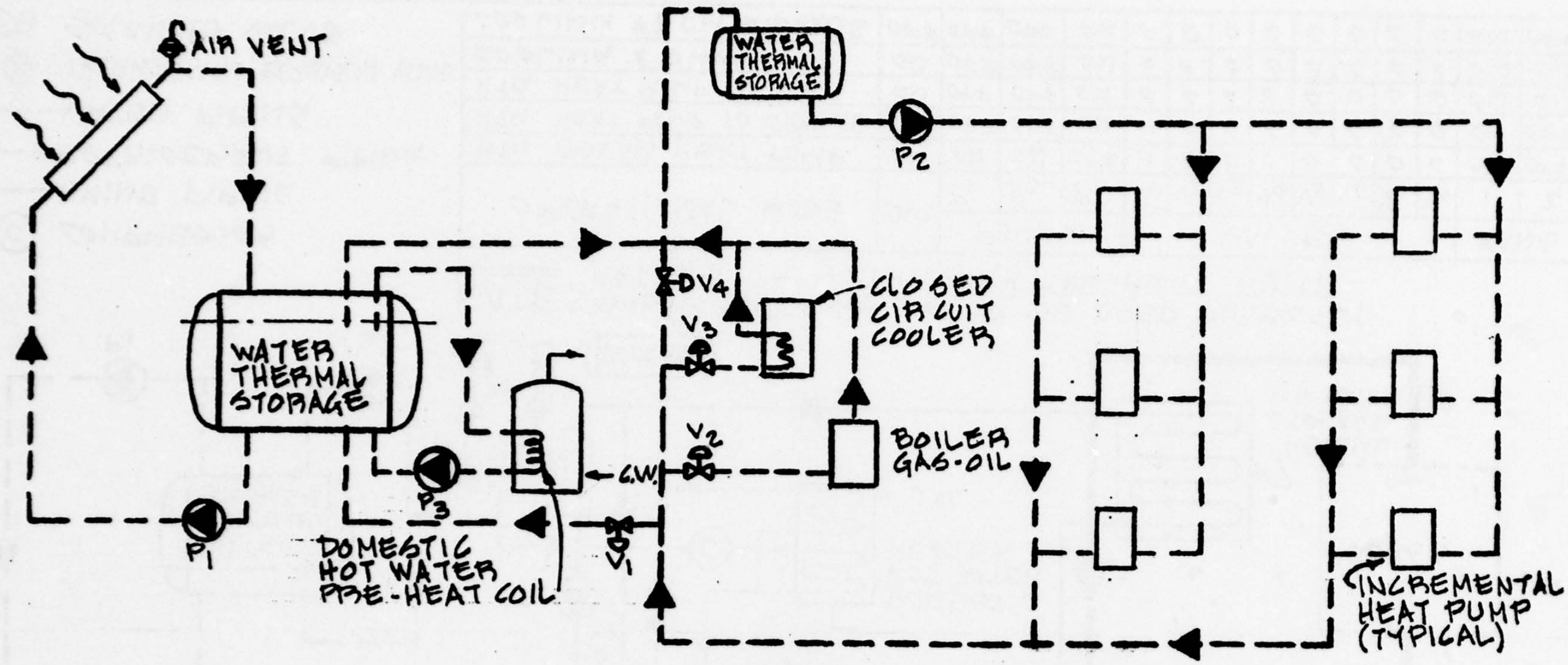
OPERATING MODE	PUMPS			VALVES				FAN
	P_1	P_2	P_3	V_1	V_2	V_3	V_4	1
HTG. SOLAR DIRECT	ON	ON	ON	O	C	C	O	ON
HTG. SOLAR HEAT PUMP	ON	ON	ON	O	C	O	C	ON
HTG. ELECT. COIL AUXILIARY	OFF	OFF	OFF	C	C	C	C	ON
SUMMER COOLING & DHW	ON	ON	ON	C	O	O	C	ON

Figure 19. System No. 17



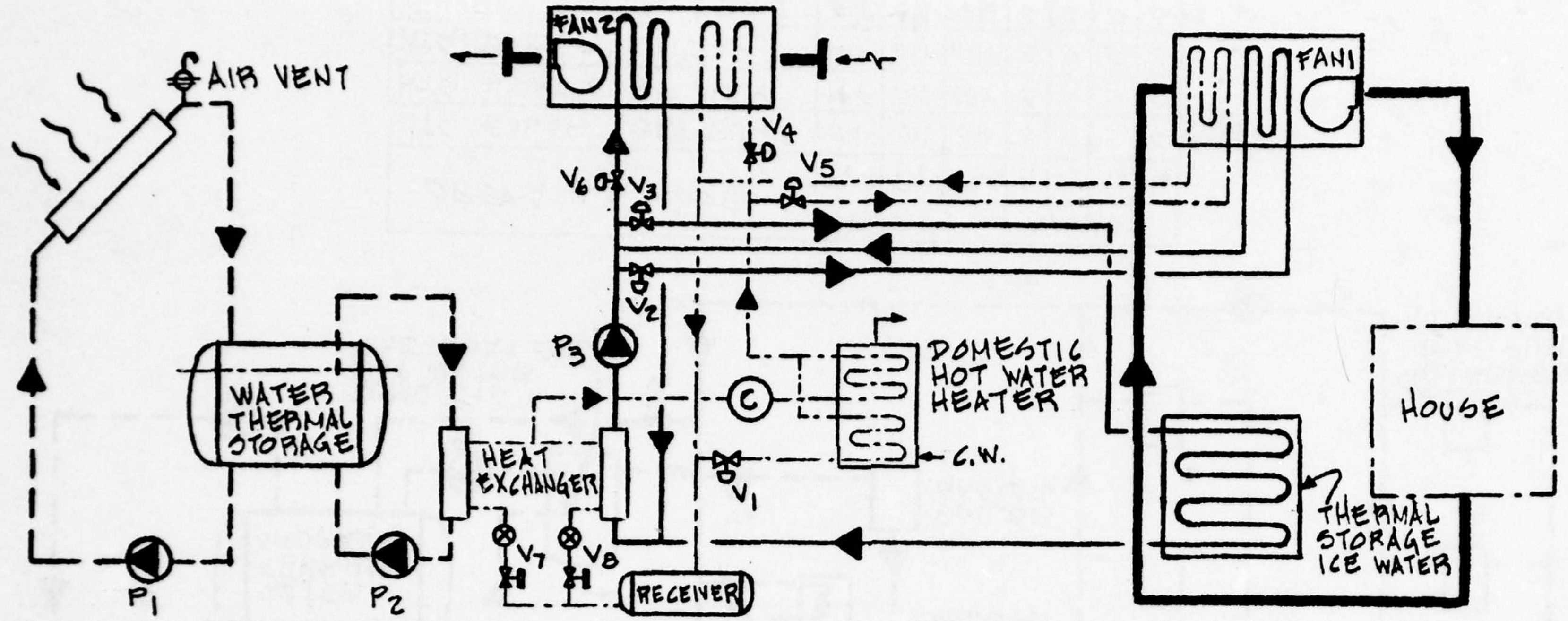
OPERATING MODE	PUMPS			VALVES				FAN
	P_1	P_2	P_3	V_1	V_2	V_3	V_4	1
HTG. SOLAR DIRECT	ON	ON	ON	O	C	C	O	ON
HTG. SOLAR HEAT PUMP	ON	ON	ON	O	C	O	C	ON
HTG. FURNACE AUXILIARY	OFF	OFF	OFF	C	C	C	C	ON
SUMMER COOLING & DHW	OFF	ON	ON	C	O	O	C	ON

Figure 20. System No. 18



OPERATING MODE	PUMPS			VALVES			
	P_1	P_2	P_3	V_1	V_2	V_3	V_4
HTG. SOLAR HEAT PUMP	ON	ON	ON	O	C	C	C
HTG. HEAT PUMP ONLY	OFF	ON	OFF	C	C	C	O
HTG. BOILER AUXILIARY	OFF	ON	OFF	C	O	C	C
SUMMER COOLING & DHW	ON	ON	ON	C	C	O	C

Figure 21. System No. 19

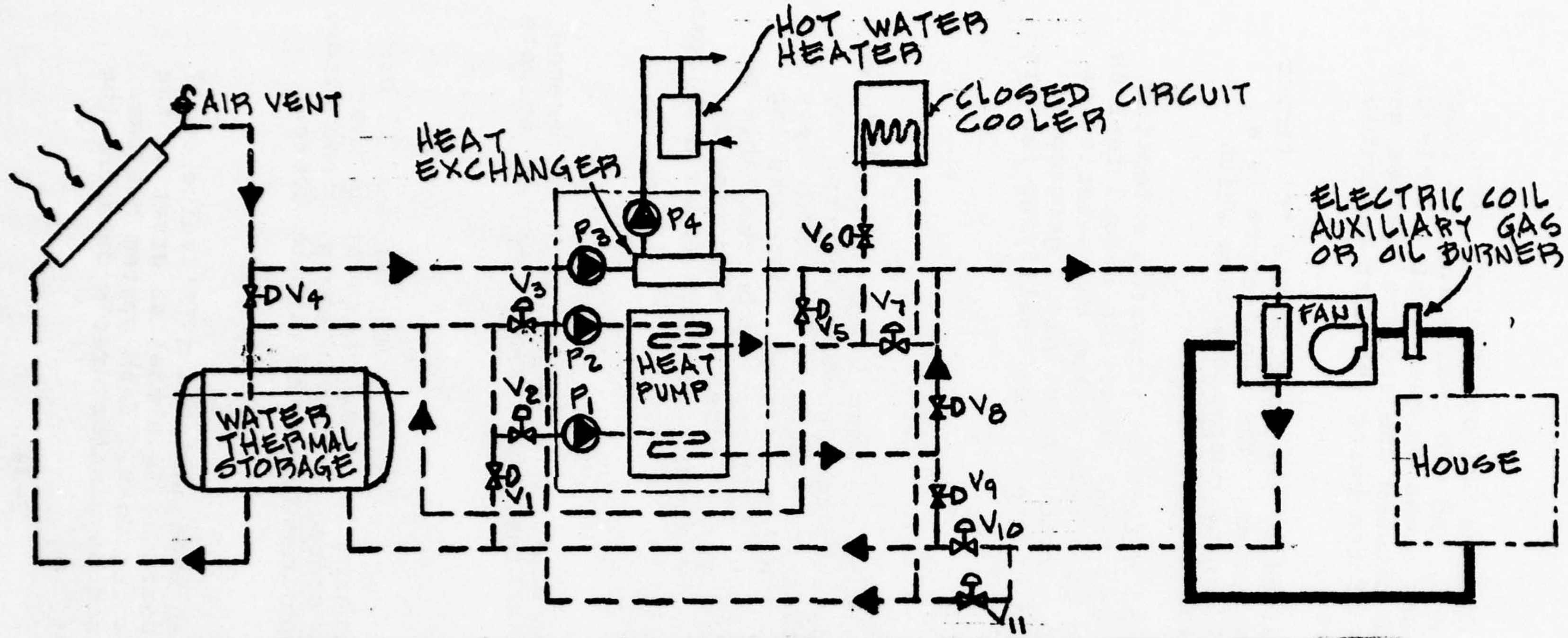


NOTE: SOLAR ENERGY COULD BE USED DIRECTLY RATHER THAN THROUGH THE HEAT PUMP.

Ⓒ COMPRESSOR
— BRINE PIPING
-·- REFRIGERANT PIPING
--- WATER PIPING
⊗ THERMOSTATIC EXPANSION VALVE
SOLENOID VALVE

OPERATIONAL MODE	COMP.	PUMPS			VALVES								FANS	
		P_1	P_2	P_3	V_1	V_2	V_3	V_4	V_5	V_6	V_7	V_8	1	2
HTG. SOLAR HEAT PUMP	ON	ON	ON	OFF	O	C	C	C	O	C	O	C	ON	OFF
HTG. HEAT PUMP TO STORAGE	ON	OFF	OFF	ON	O	C	O	C	O	C	C	O	ON	OFF
HTG. HEAT PUMP TO O.A.	ON	OFF	OFF	ON	O	C	C	C	O	O	C	O	ON	ON
COOLING & DHW	ON	OFF	OFF	ON	O	O	O	O	C	C	C	O	ON	ON
COOLING FROM STORAGE	OFF	OFF	OFF	ON	C	O	O	C	C	C	C	C	ON	OFF

Figure 22. System No. 20



OPERATING MODE	PUMPS				VALVES											FAN
	P_1	P_2	P_3	P_4	V_1	V_2	V_3	V_4	V_5	V_6	V_7	V_8	V_9	V_{10}	V_{11}	1
HTG. SOLAR DIRECT	OFF	OFF	ON	ON	C	C	C	C	C	C	C	C	C	O	C	ON
HTG. UPGRADING SOLAR DIRECT	ON	ON	ON	ON	C	O	C	C	O	C	O	C	O	C	O	ON
SOLAR STORAGE & DHW	OFF	OFF	ON	ON	C	C	C	C	O	C	C	C	C	C	C	OFF
HTG. STORED SOLAR	OFF	OFF	ON	ON	C	C	C	O	C	C	C	C	C	O	C	ON
HTG. UPGRADING STORAGE	ON	ON	OFF	OFF	C	O	C	C	C	C	O	C	O	C	O	ON
COOLING	ON	ON	OFF	ON	O	O	C	C	C	O	C	O	C	C	C	ON

Figure 23. System No. 21

by assigning a 'K' factor between one and twenty; twenty indicating the most important, and one indicating the least important. These factors are shown on the Matrix, Table 2. Each system was then scored between one and ten against each qualifying factor, a score of ten being the most desirable and one the least desirable.

Each score was multiplied by the appropriate 'K' factor and then totalled for each system. The final score is a combination of the overall system characteristics which gives an indication of priority.

Although this is the main factor for system selection, it is not the only factor and final selection should include consideration and matching of unique system characteristics with the requirements of a particular building, geographic location and climate. Each of the system qualifying factors is discussed below:

Availability

This is considered one of the most important qualifying factors. When making judgments against availability, the highest score indicates that the equipment is immediately available off the shelf. The lower scores, in descending order, indicate that equipment will shortly be available or will be available in 3 to 4 years' time. The scoring for different systems will change as equipment now under development becomes commercially available.

First Cost

This factor has been assigned a 'K' value of 18 because its relative importance is high. High system scores indicate low cost and vice versa.

Operating Cost

Operating Cost has been assigned a 'K' value of 17 because its relative importance is almost as great as the First Cost. Systems are scored on yearly energy consumption, the most efficient systems receiving the highest scores.

Maintenance Cost

Maintenance Cost has been assigned a 'K' value of 17 because its relative importance is almost as great as the First Cost and the Operating Cost. Each system has been scored subjectively, based on manufacturer's data and the contractor's experience.

Life Cycle Cost

The Life Cycle Cost is a combination of First Cost, Operating Cost, and Maintenance Cost, and has been assigned a 'K' value of 17. When making judgments against this factor, systems that have the lowest First Cost, are easiest to maintain, and have high COPs, received the highest marks.

Space Requirements

This factor has been assigned a 'K' value of 12 because it is relatively important in retrofit situations. Generally, air collector systems with rock storage and air ducts connecting the various pieces of equipment required more space than equivalent water systems and have been marked down accordingly.

Serviceability

This factor has been assigned a 'K' value of 10. Serviceability has been defined as "ease of access to important components for maintenance" and little difference was found in any of the proprietary machines, although, in general, split systems were easier to service than unitary systems. This is reflected in the slight variation in scoring.

Useful Life

This factor was assigned a 'K' value of 16 because it was felt to be one of the more important factors to be considered when selecting a given system. Generally, conventional systems that have been marketed for a number of years, gave relatively trouble-free performance with an average useful life of 15 years. Systems which were more experimental in nature and which had no operating history, were felt to be more risky and have been marked down accordingly.

Seasonal Performance

This factor has been assigned a 'K' value of 15 and is relatively important as it directly affects the operating cost of the system. Seasonal performance is the yearly energy consumption of a system to meet given heating and cooling loads. It also reflects the percentage of solar participation on a yearly basis. Systems that use solar assistance for heating only have been given a lower mark than systems which can use solar energy for both heating and cooling.

Noise

This factor was assigned a 'K' value of 5. It is considered relatively unimportant because most heat pump systems presently marketed are either quiet in operation or have been acoustically treated within the unit. Split systems are inherently less troublesome because the compressor and condenser units are mounted outside the house.

Reliability

This factor was assigned a 'K' value of 14 because it was judged to be relatively important. Systems employing conventional heat pumps in conjunction with air collectors were judged to be the most reliable, followed by conventional heat pumps combined with liquid collectors. Unconventional and experimental systems were judged to be poor risks and have been marked down accordingly.

Climatic/Geographic Suitability

This factor was assigned a 'K' value of 3 because all the systems reviewed can be adapted for use with varying degrees of success in any climatic/geographic location within the Continental United States. Systems employing air collectors require the least modification to counter climatic effects and were marked highest. Systems employing liquid collectors must be protected from frost damage in areas subject to freezing temperatures and were marked down. System 15, which employs water as a heat sink/source, was marked down because it can only be used in a geographic location where a large body of water is available. System 20 was marked down because it requires a balanced heating and cooling load for optimum yearly operation.

Control Complexity

This factor was assigned a 'K' value of 13. Simple systems with few control valves or dampers were given the highest marks. More complex systems were marked down accordingly.

Crankcase Heater

This factor was assigned a 'K' value of 4. Early material suggested that tangible savings of electrical energy could be made by eliminating or turning off crankcase heaters. Further investigation with both manufacturers and the maintenance chief at Little Rock Air Force Base indicated that energy used by crankcase heaters

is relatively small (50 Watts/machine) and that elimination of heaters could cause serious damage to compressors on start-up.

Freeze Protection

This factor was assigned a 'K' value of 8. Systems employing air collectors were given the highest marks because they require little or no protection against frost damage. Systems employing liquid collectors were marked down, in proportion.

Peak Electrical Demand

This factor was assigned a 'K' value of 14. Systems capable of both heating and cooling with solar power and which require electrical input only as auxiliary backup were given the highest marks. Systems that require electrical energy input for both heating and cooling were marked down because there is a clear probability that a number of these systems operating together under peak load conditions could create a peak electrical demand.

Leaks

This factor was assigned a 'K' value of 6 because both pipework and ductwork installed correctly, by definition, don't leak. Systems are then marked according to the impact of leaks upon the system operation and the effect of the leakage on the building. Air systems are more leaky than water systems but the effect of small air leaks from ducts is negligible and will not cause damage to the building. Leaks from water systems, however, will tend to interfere with the operation of the equipment and could cause expensive damage within the building. Water systems have been marked down accordingly and are graded according to the length of pipework installed and potential for leaks.

Corrosion

This factor was assigned a 'K' value of 10. Systems employing air collectors and air storage are given the highest marks as the likelihood of corrosion within the collectors, ducts, and concrete bin is very low. Systems employing liquid collectors and liquid storage are more susceptible to corrosion, particularly if dissimilar metals are used (aluminum absorber plates) and these systems have been marked down accordingly.

Effects on Storage

This factor was assigned a 'K' value of 14 because storage is an indispensable part of any solar assisted heat pump system. Systems were graded on the relative volume of storage required and the ease with which it could be accommodated, either within or outside the building. Generally, systems employing liquid storage received the highest marks and systems employing air storage were marked down accordingly. Exceptions to this are Systems 11, 12, and 13, which use liquid storage, but require high storage temperatures in the summer for operation in the cooling mode, with attendant expansion and pressurization requirements. These three systems were downgraded because their effect on storage will be greater than any other system.

Effects on Collectors

This factor was assigned a 'K' value of 12. Systems that do not require a complicated high efficiency collector are given the highest marks. Systems 11, 12, and 13, which require high collection temperatures in summer for satisfactory operation, were marked down because these systems require more expensive collectors that will operate efficiently at high temperatures.

Effects on Building Structure

This factor was assigned a 'K' value of 20 and, together with Availability, is the most important consideration when selecting systems. Systems were rated on the amount of space required for installation and the modifications that must be made to the building structure to allow the installation to take place. Split heat pump systems employing liquid collectors and storage were given the highest marks. Air collectors require more alteration to the structure and larger openings to accommodate ductwork, and were marked down accordingly. Systems 7 and 8, which employ attic collectors, would require complete re-roofing of the house and have received a low mark. The lowest rating was given to System 20, which would require extensive building modifications to accommodate the seasonal ice storage tank.

Visual Intrusion

This factor was assigned a 'K' value of 4 because it is felt to be a relatively unimportant consideration when selecting a system. Liquid systems were given the highest marks and air systems have been down-graded because it is usually easier to incorporate pipework than ductwork.

Potential for Local Zone Control

This factor was assigned a 'K' value of 10 and is of medium importance because zone control would allow unoccupied areas to be isolated, thus reducing the load. All of the systems except 19 and 21 have very limited potential for zoning control. Systems 19 and 21, which use liquid as the distributing medium to terminal units, have good potential for zone control and have been given the highest marks.

Aesthetic Appearance

This factor was assigned a 'K' value of 12 because the systems should not destroy the architectural appearance of the house. The most visible component of the systems is the solar collector. Water collectors are usually neater in appearance and easier to integrate architecturally than air collectors, although both types of collector are surface-mounted on the roof. The best collector from an aesthetic point of view is the attic type, where the only evidence visible from the outside is a series of windows let into the roof having the appearance of a skylight. This system received the highest mark for appearance, with liquid and air systems being downgraded respectively.

Safety

This factor was assigned a 'K' value of 16 because it is obviously a major consideration. Systems employing air collectors and rock storage are inherently safer than liquid systems which are subject to freezing in extremely cold weather, and over-pressure during periods of high temperatures and high insolation rates.

2.8 SYSTEM SELECTION

As a result of the matrix evaluation, each of the 21 systems has a total evaluation score. Table 3 tabulates the systems in order of rank from the highest to the lowest.

The first six systems were tentatively selected for further evaluation. However, each of the systems has characteristics which can make it more or less desirable in a given geographic location and climate, and for existing conditions at the selected location.

TABLE 2. SYSTEM ANALYSIS MATRIX

'K'	QUALIFYING FACTORS	SYSTEM DESIGNATION																				
		AIR/AIR													WATER/AIR							WATER/WATER
		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
20	... Availability	10	10	10	10	10	10	10	10	7	7	3	5	3	10	10	10	10	8	10	7	10
18	... First Cost	10	6	10	7	5	8	8	9	6	6	1	1	1	6	4	3	8	7	5	3	6
17	... Operating Cost	6	6	6	6	6	6	6	6	8	8	10	10	10	7	9	8	6	7	9	7	5
17	... Maintenance Cost	9	8	9	9	8	9	10	10	4	4	2	2	2	9	8	8	10	9	8	7	7
17	... Life Cycle Cost	10	8	10	9	8	9	10	10	7	7	5	5	5	9	8	8	10	9	9	7	6
12	... Space Requirements	7	9	8	7	10	8	7	7	7	9	8	8	8	8	9	9	10	8	9	5	8
10	... Serviceability	9	9	9	10	10	10	8	8	9	9	9	9	9	10	7	7	10	10	10	8	8
16	... Useful Life	10	9	10	10	9	10	10	10	9	7	7	9	8	9	7	7	10	9	10	9	10
15	... Seasonal Performance.....	7	6	7	7	6	7	7	7	8	8	9	9	9	7	9	8	6	7	9	10	7
5	... Noise	9	9	9	10	10	10	10	9	7	7	7	6	7	8	8	8	9	8	8	10	9
14	... Reliability	10	9	10	10	9	10	10	10	7	6	2	6	4	9	8	8	10	9	9	8	8
3	... Climatic/Geographic Suitability	10	8	8	10	8	8	10	10	10	8	8	8	8	8	5	7	8	8	8	6	8
13	... Control Complexity	9	8	10	9	8	10	10	10	9	8	7	7	7	9	10	10	9	9	10	8	7
4	... Crankcase Heater	9	9	9	8	8	8	9	9	10	10	10	10	10	8	8	8	8	8	8	8	8
8	... Freeze Protection	10	7	10	10	7	10	10	10	10	7	7	7	7	7	7	7	7	7	7	7	7
14	... Peak Electrical Demand ..	7	7	7	7	7	7	7	7	7	7	10	10	10	8	8	8	7	8	9	10	7
6	... Leaks	10	8	10	10	8	10	9	9	10	8	8	8	8	10	10	8	8	10	10	8	8
10	... Corrosion	10	6	10	10	6	10	10	10	10	6	6	6	6	5	4	4	6	5	6	4	6
14	... Effects On Storage	8	10	8	8	10	8	8	8	8	10	7	7	7	9	10	10	9	9	8	4	9
12	... Effects On Collector	9	9	9	9	9	9	10	10	9	9	7	7	7	9	9	9	9	9	9	9	9
20	... Effects On Building Structure (Retrofit)...	7	9	9	8	10	9	3	3	8	10	10	10	10	9	7	7	10	9	10	1	9
4	... Visual Intrusion Into Occupied Areas	8	10	9	8	10	9	8	8	8	10	10	10	10	10	10	10	10	10	9	10	10
10	... Potential For Local Zone Control	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	10	4	9
12	... Aesthetic Appearance	7	9	7	9	7	9	10	10	7	9	9	9	9	9	9	9	9	9	9	9	9
16	... Safety	10	8	10	10	8	10	10	10	10	8	8	8	8	7	8	8	8	8	7	6	8
		2626	2455	2689	2610	2456	2671	2577	2590	2363	2314	2020	2143	2064	2509	2423	2367	2626	2503	2665	2053	2401

Table 3. Table of Systems In Order of Rank

System Number	Evaluation Score	Rank Order	Heat Pump Type	Collector Type
3	2689	1	Air/Air	Air
6	2671	2	Air/Air	Air
19	2665	3	Water/Air Incremental	Liquid
17	2626	4	Water/Air	Liquid
1	2626	5	Air/Air	Air
4	2610	6	Air/Air	Air
8	2590	7	Air/Air	Attic
7	2577	8	Air/Air	Attic
14	2509	9	Water/Air	Liquid
18	2503	10	Water/Air	Liquid
5	2456	11	Air/Air	Liquid
2	2455	12	Air/Air	Liquid
15	2423	13	Water/Air	Liquid
21	2401	14	Water/Water	Liquid
16	2367	15	Water/Air	Liquid
9	2363	16	Rovac	Air
10	2314	17	Rovac	Air
12	2143	18	Rankine	Liquid
13	2064	19	Stirling	Liquid
20	2053	20	Water/Air	Liquid
11	2020	21	Pivoting Tip Vane	Liquid

a. System 3. This system employs an air collector and rock storage, and can be used in conjunction with any existing heat pump, either split or unitary. The solar collection system will operate independently from the heat pump and when enough solar energy is available, will be used to heat the house with excess heat being used to charge the storage bin. When solar heating is not possible and the thermal storage bin has been exhausted, the heat pump system will operate in its normal manner. This system is not strictly a solar assisted heat pump, but uses solar energy in place of the heat pump and reduces the operating hours of the heat pump with a consequent saving of electrical energy. Research indicates that in areas where the winter design temperature is greater than 20°F, only small energy savings can be achieved by improving the heat pump COP with solar pre-heat. In locations experiencing this climate it is not cost effective to install a more complicated system. If, however, the location under consideration is subject to extreme temperatures in winter (10°F and lower), then using the solar energy system to improve the COP becomes more economically attractive and a different type of system will take precedence.

b. System 6. System 6 is similar to System 3, but employs a liquid collector and water storage instead of an air collector and rock storage. Apart from this difference the operation of both systems is identical, as are the characteristics which make them suitable for some geographic locations, but not others.

c. System 19. This system employs centralized liquid collectors and water thermal storage to heat a common hydronic loop serving a group of houses. Each house is provided with one or more incremental water-to-air heat pumps which are connected to the common loop. In the heating mode when solar energy is available, it is used to add heat to the loop, the heat being removed from the loop by the individual heat pumps and upgraded to meet the heat load of the houses. In the cooling mode the heat pumps cool the houses and reject heat into the common loop which, in turn, is cooled by a central closed-circuit cooler. In spring and fall when there is likely to be a period of time when cooling is required during the day and heating at night, the heat rejected into the loop during the day is stored in a large water thermal storage tank and used as a heat source for the heat pumps at night. Thus, on a daily basis, no heat energy is added to or taken from the system. The disadvantage of this system is that it can be used only for large groups of houses. It does, however, have the advantage of using a low

grade heat source which means that the solar collector system can operate at relatively low temperatures and very high collection efficiencies. It also possesses the capability of short-term energy storage and reuse. The system can be used in any geographic location and climatic area, but would be most favorable in areas that experience balanced heating and cooling loads.

d. System 17. This system employs liquid solar collectors with water thermal storage feeding a unitary water-to-air heat pump. When it is possible to collect solar energy at temperatures exceeding 100°F, the solar heated water will be circulated through a coil in the supply air ductwork to the house, and the solar energy used directly to meet the heating load. In this mode of operation the heat pump will be turned off and only the supply fan used. When it is not possible to collect solar energy at direct utilization temperatures, but it is possible to collect at temperatures in excess of 60°F, then the solar heated water will be used as a heat source for the heat pump. After a period of little or no insolation and when the heat in the thermal storage tank has been depleted (water temperatures less than 45°F) the heat pump will be turned off and the auxiliary heating coil used to meet the load. This system has the advantage of simplicity and positive operation, but has the disadvantage of poor yearly performance in areas of low winter insolation when the auxiliary back-up system must be used for long periods of time. Its selection should, therefore, be limited to areas enjoying high levels of winter insolation or where alternative natural water heat sources such as lakes or well water are available.

e. System 1. This system employs air collectors and rock storage in conjunction with an air-to-air unitary heat pump. Whenever it is possible to collect heat at temperatures greater than 80°F, solar energy will be used directly to meet the heating load and the heat pump will be turned off. During periods of low insolation when collection temperatures are too low for direct use, but are higher than ambient air temperatures, the solar heated air will be used as a heat source for the heat pump which will then operate at a higher COP than if it used the colder outdoor air as a heat source. During periods when solar collection is not possible and the heat in the rock storage is depleted, the air-to-air heat pump will revert to its normal operation providing the outdoor air temperature is sufficiently high to enable the heat pump to meet the heating load. When solar heat is not available and the outdoor temperature is too low for satisfactory heat pump operation, auxiliary heating will be used.

This system, although more expensive and complex than Systems 3 and 6, is more versatile because it can be used in cold climates and a wider variety of geographic locations.

f. System 4. System 4 is identical to System 1, except that it uses a split air-to-air heat pump, instead of a unitary heat pump. It has the same advantages and disadvantages as System 1, but is a little more complex because air must be ducted to the outdoor portion of the heat pump and an extra fan must be provided to insure positive air circulation through the outside coil. In a retrofit situation where the outdoor section of a split heat pump is located remote from the house, this system might be precluded from use on the grounds of difficulty and cost of running the outdoor ductwork. It could, however, be used if it is possible to relocate the outdoor unit near a house in a protected position such as a carport or outdoor storage shed.

As discussed above, some of the six selected systems, while being excellent choices for a given location and set of conditions, may not be suitable in other areas where less favorable conditions exist. In this case, alternative candidate systems which suit the peculiarities of a given location or set of conditions should be chosen from the remaining 15 systems.

Systems 8 and 7 are next in order of rank and are both attic collector systems employing air-to-air heat pumps. These systems would be particularly suitable in a retrofit situation where the roof of the house had reached the end of its life and was shortly to be replaced. In this case, the roof could be redesigned and replaced to incorporate the attic collector system at very little extra cost over replacing it with regular roofing materials. If, however, the roof of the house is on a relatively shallow slope and will not need replacing for some years, the cost of incorporating an attic air collector system would preclude the use of these systems.

2.9 ANALYTICAL DISCUSSION

In order to determine the effect of a solar assisted heat pump system on annual energy consumption, an analysis of a hypothetical residence was done. A detailed analysis of the actual system and residences involved was carried out for the second phase of this study, however, the results and conclusions summarized in this section are sufficiently accurate to estimate the order of magnitude impact of a solar assisted heat pump system.

The following basic assumptions were made for the analysis:

a. Latitude 32-37°N.

b. Heating degree days (base 65°F) - 3300.

c. Cooling degree hours (base 75°F) - 23785.

d. Winter design temperature (peak) - 20°F.

e. Summer design temperature (peak) - 96°F.

f. Domestic hot water load - 25 gallons per person per day from 55-140°F, generation efficiency - 75 percent.

g. Residence floor area - 1500 sq ft.

h. Number of occupants - 4.

i. Heating load - 7.5 Btu/sq ft per degree day (base 65°F).

j. Peak heating load based on above - 21,094 Btu/hr.

k. Cooling load - 1.3 Btu/sq ft per degree hour (base 75°F).

l. Peak cooling load based on above - 40,950 Btu/hr.

m. Collector tilt - 45° from horizontal.

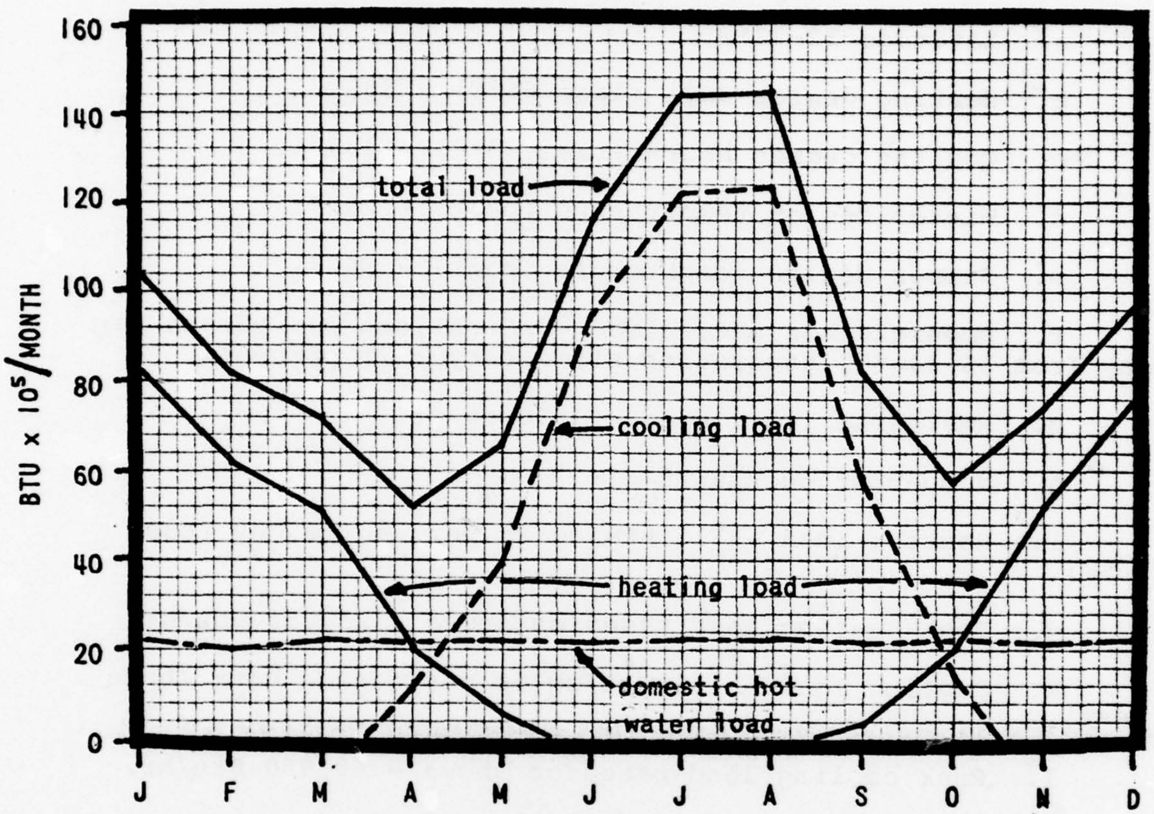
n. Collector outlet temperature - 120°F (October-April), 140°F (May-September).

o. Collector efficiency - (120-140°F range) based on typical liquid collectors such as Ametek, International Environment Corp., Honeywell, Chamberlain.

p. Heat pump performance - based on typical state-of-the-art, air-to-air heat pumps - Westinghouse HI-RE-LI split system.

Weather data were taken from AFM 88-8 (Engineering Weather Data) and the Climatic Atlas of the United States. It was assumed that heating would be required for outdoor temperatures below 65°F and that cooling would be required for outdoor temperatures above 75°F.

Figure 24 shows graphically the magnitude and annual distribution of the heating, cooling, and domestic hot water



1,500 sq. ft. residence
heating degree days (base 65°F.) 3,300
cooling degree hours (base 75 F.) 23,785
peak heating load 21,094 btu/hr.
peak cooling load 40,950 btu/hr.

Figure 24. Annual Load Distribution

loads. Figures 25 and 26 represent heating and cooling season heat pump performance respectively. Figure 25 shows how the effect of solar assist increases drastically for outdoor temperatures below approximately 15°F. In geographical areas with mild winters such as those with load distributions similar to Figure 24 where there are few hours at temperatures less than 15°F, increasing heat pump COP with solar energy will not be economically viable. The increase in COP due to the solar system is not great enough for temperatures above 15°F to offset the additional costs involved. To use solar energy to raise the COP of an air to air heat pump, additional ductwork, automatic dampers and controls are required at substantial extra cost. This expense can generally be justified for latitudes above 40°N. In milder climates such as the hypothetical location analyzed, it would be more effective to install a solar energy system for direct heating and domestic hot water only and let the heat pump operate independent of the solar system.

A graphic comparison of a conventional heat pump system and a solar assisted heat pump system is shown in Figures 27 and 28. The monthly differences in magnitude show the effect of the solar system on energy input. Collector system output includes an allowance for cloudy days based on cloud cover percentages taken from the Climatic Atlas and is also based on a collector area of 200 sq ft. This system will result in a solar participation of approximately 60 percent of the annual heating and domestic hot water loads. The amount of solar participation could be increased by installing more collectors, however, the most cost effective participation is in the 55-70 percent range.

Figure 29 is derived from Figures 27 and 28 to show the actual amount of solar energy used for the hypothetical system under consideration. The percentages shown for each month indicate the portion of the total heating and domestic hot water loads met by solar energy. This distribution is typical for the most cost effective use of solar energy.

In locations where the cooling load is dominant, say below 35°N, the effect of a solar system on annual energy consumption would be increased significantly if solar energy were also used to meet the cooling load. Accordingly, the present day status of solar assisted cooling systems was investigated. Solar assisted absorption chillers are available but not in a heat pump configuration. Consequently, this system was ruled out. The only viable alternative is a turbine/compressor using a Rankine cycle for the turbine drive. The Hamilton Standard Division of United Technologies

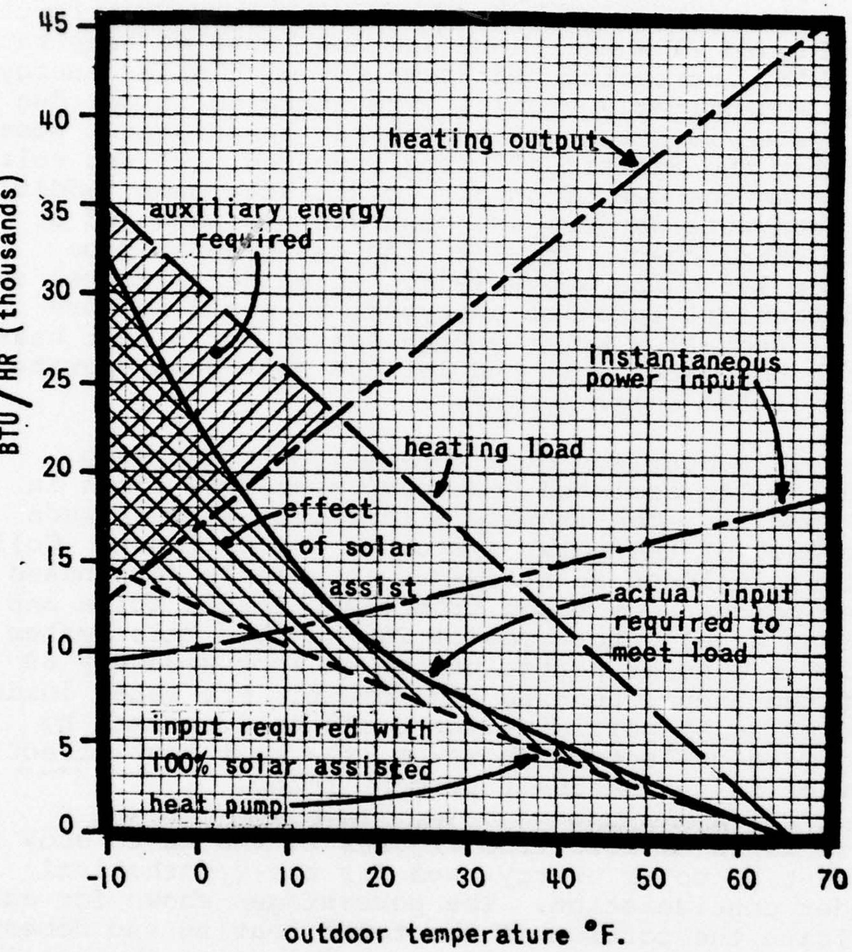


Figure 25. Typical Heat Pump Heating Season Performance

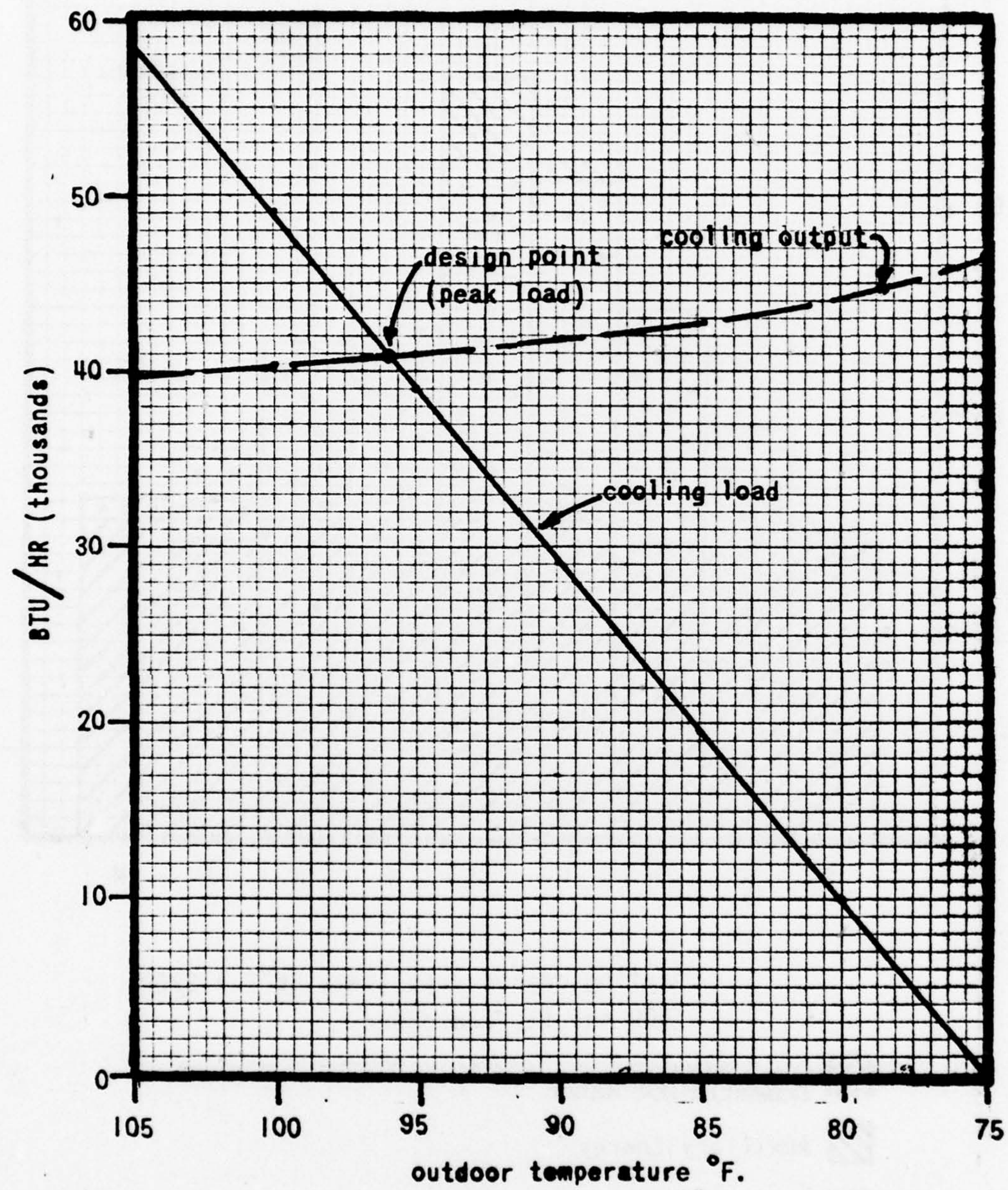


Figure 26. Typical Heat Pump Cooling Season Performance

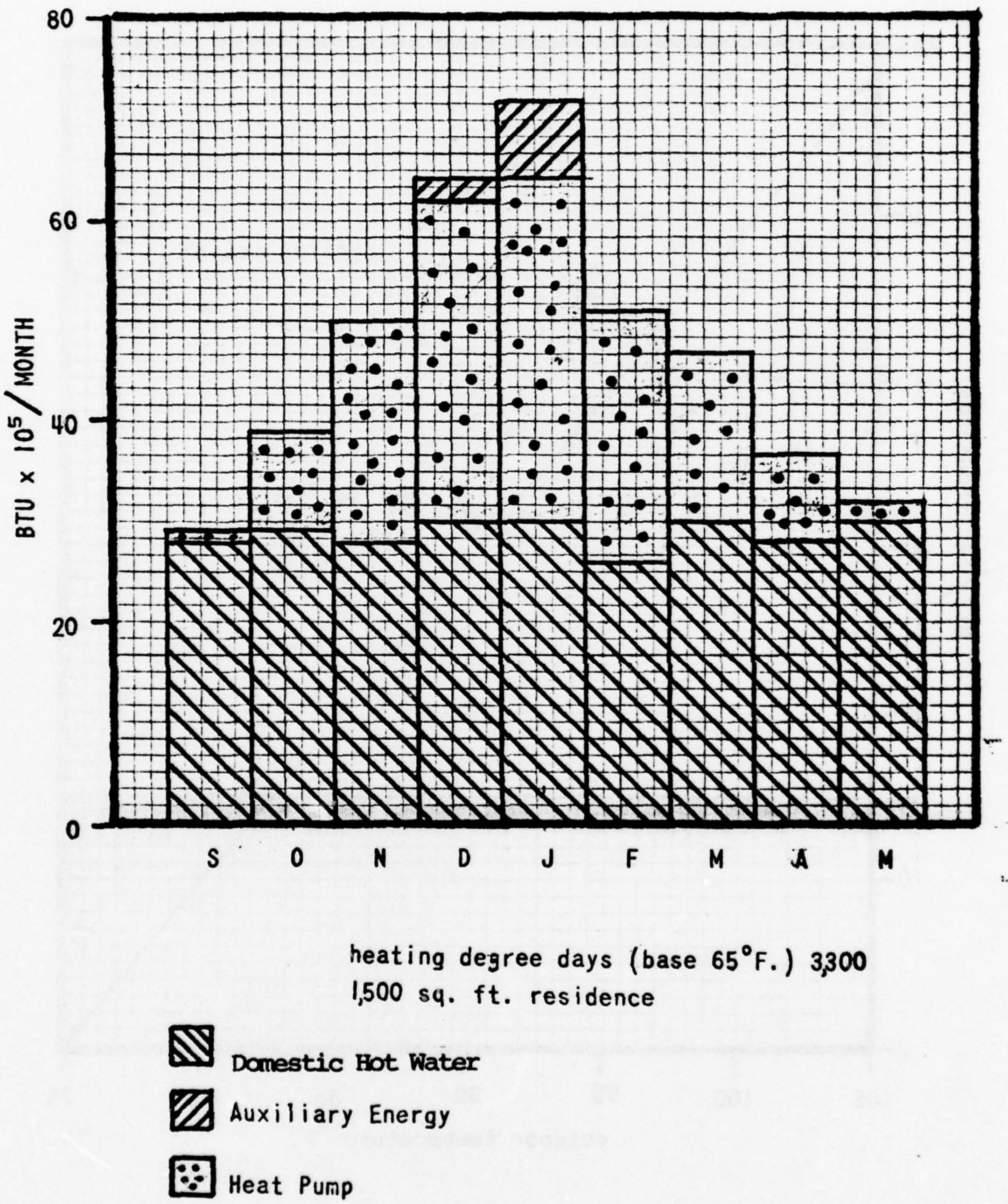
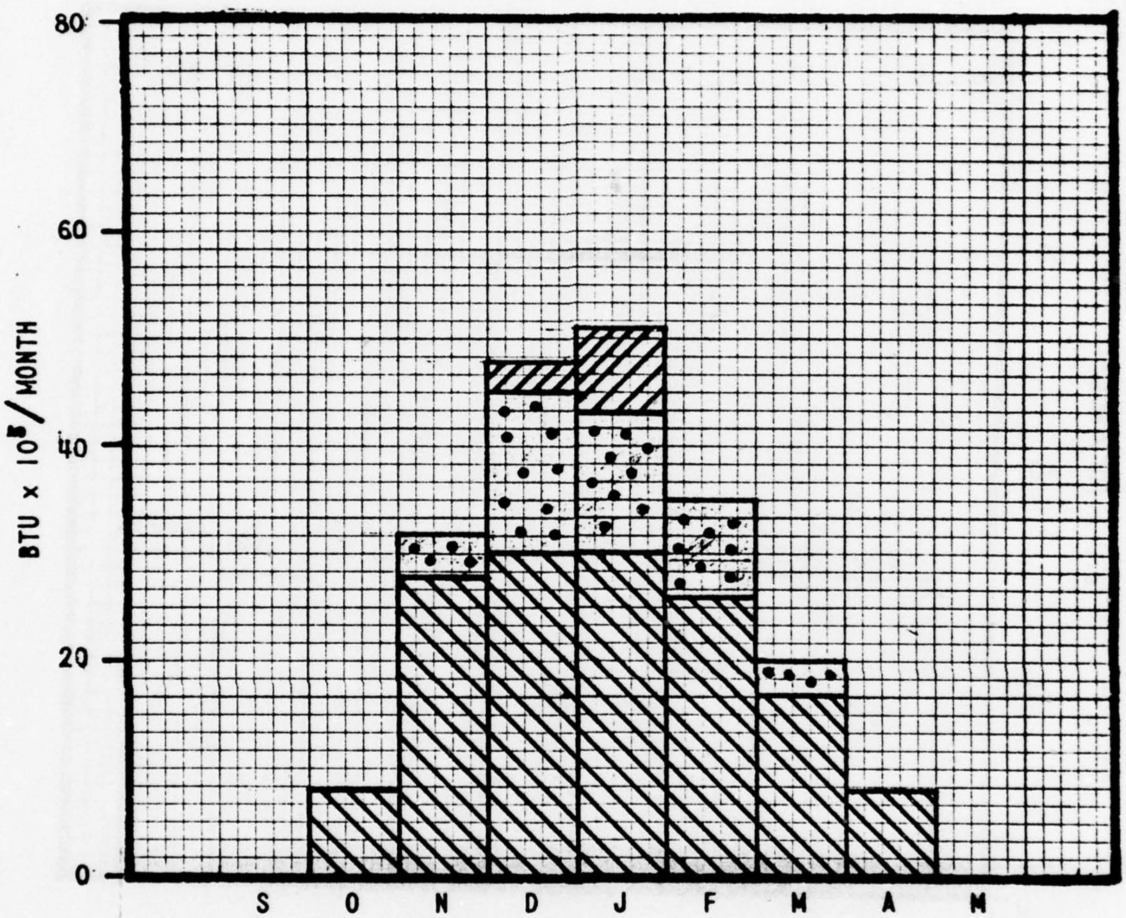


Figure 27. Heating Season Energy Input
Conventional Heat Pump System



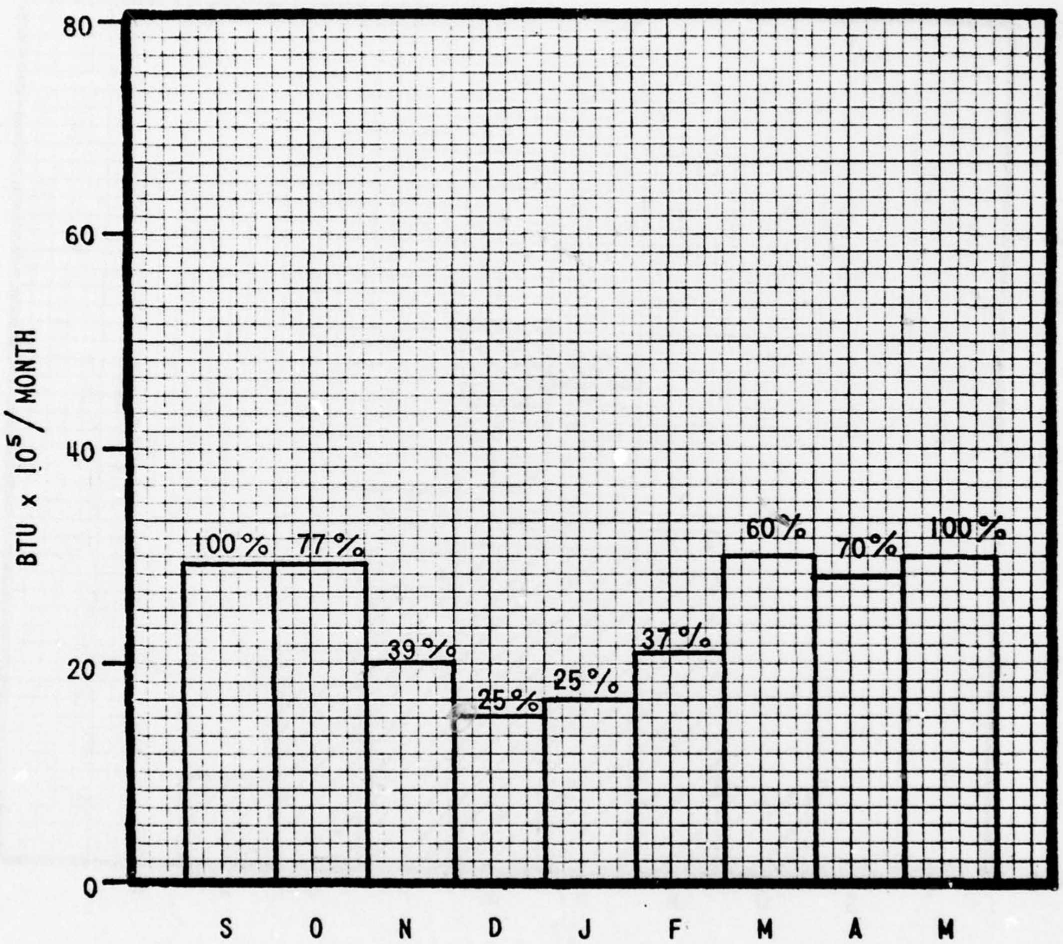
heating degree days (base 65°F.) 3,300
1,500 sq. ft. residence
collector area+200 sq. ft.

Domestic Hot Water

Auxiliary Energy

Heat Pump

Figure 28. Heating Season Energy Input
Solar Assisted Heat Pump System



NOTE: percentages indicate the ratio of solar energy utilized-to-the energy input required to meet heating and domestic hot water loads.

heating degree days (base 65°F.) 3,300
1,500 sq. ft. residence
collector area=200 sq. ft.

Figure 29. Solar Energy Utilization - Heating Season

is presently working on a drive of this type and has prototype models operating with input temperatures in the 160-170°F range. This system has the best potential for maximum utilization of solar energy, and is worthy of further study and development effort. However, commercialization of this system is at least three years away.

2.10 FUTURE DEVELOPMENTS

While compiling material for this report, it became evident that there were several systems under development which show great potential for future use. These systems are the Rovac air cycle, the low temperature Rankine turbine heat pump, and the Stirling engine heat pump. Of the three, the Stirling heat engine is the least developed and present research is directed mainly at producing this machine in the form of an alternative drive for systems normally using electric motors. The engine employs external combustion as its motive power and is extremely quiet and vibration-free in operation. At the present it can be made to operate satisfactorily with high temperature energy input (approximately 600°F) but low temperature operation is largely experimental. This alternative method of drive is about 5 years away from commercial application and even then it may be limited only to high temperature energy input.

On the other hand, the Rovac air cycle heat pump and the Rankine heat pump are both in the pilot stage of development and close to commercial exploitation. The Rovac air cycle heat pump retains conventional electric drive but has the advantage of much higher COPs at very low ambient air temperatures when compared with present generation heat pumps. This machine will probably be marketed in the near future and, if funds permit, should be considered as a demonstration unit in one house.

The low temperature Rankine turbine heat pump is currently being developed by a number of companies including General Electric, Honeywell/Lennox, and Hamilton Standard. Of all the developing machines, this heat pump shows the greatest potential in improving yearly performance as it uses solar energy both as a motive force and as a heat source, thus enabling solar participation in both cooling and heating. As with the Rovac air cycle machine, if funds permit, it is recommended that a low temperature Rankine turbine heat pump system be installed in one house as a pilot system, both to speed its development and commercial acceptance, and to gain experience in its operation.

SECTION III

PHASE II

3.1 GENERAL

To make a comparative analysis of the two most promising systems selected during Phase I, it was necessary to adopt a common base. It was recognized that either system may in actual fact be installed in houses of various designs, condition, orientation, and geographic location.

For the purpose of this report, heating and cooling loads were calculated for a typical single family, ranch-type house (see Figure 30) located in Little Rock, Arkansas, and were used to judge comparative system performance.

This typical house was also used to illustrate the constraints imposed on system installation by an existing structure. These constraints fall broadly into two categories, i.e., location of major components and effect of orientation on collector location options.

The two systems selected for this phase of the study were System 3 (Figure 5) and System 6 (Figure 8).

3.2 OBJECTIVES

The objectives of Phase II were:

- To develop in more detail and to refine the two selected systems.
- To make tentative selections of system components.
- To estimate the construction costs of comparable size systems.
- To determine the performance effectiveness of each system when meeting a given annual load.
- To determine the overall economics and comparative cost effectiveness of each system.

3.3 METHODOLOGY

The methodology in compiling this phase of the report was first to establish a typical representative building with a split system heat pump for heating and cooling (see Figure 30).

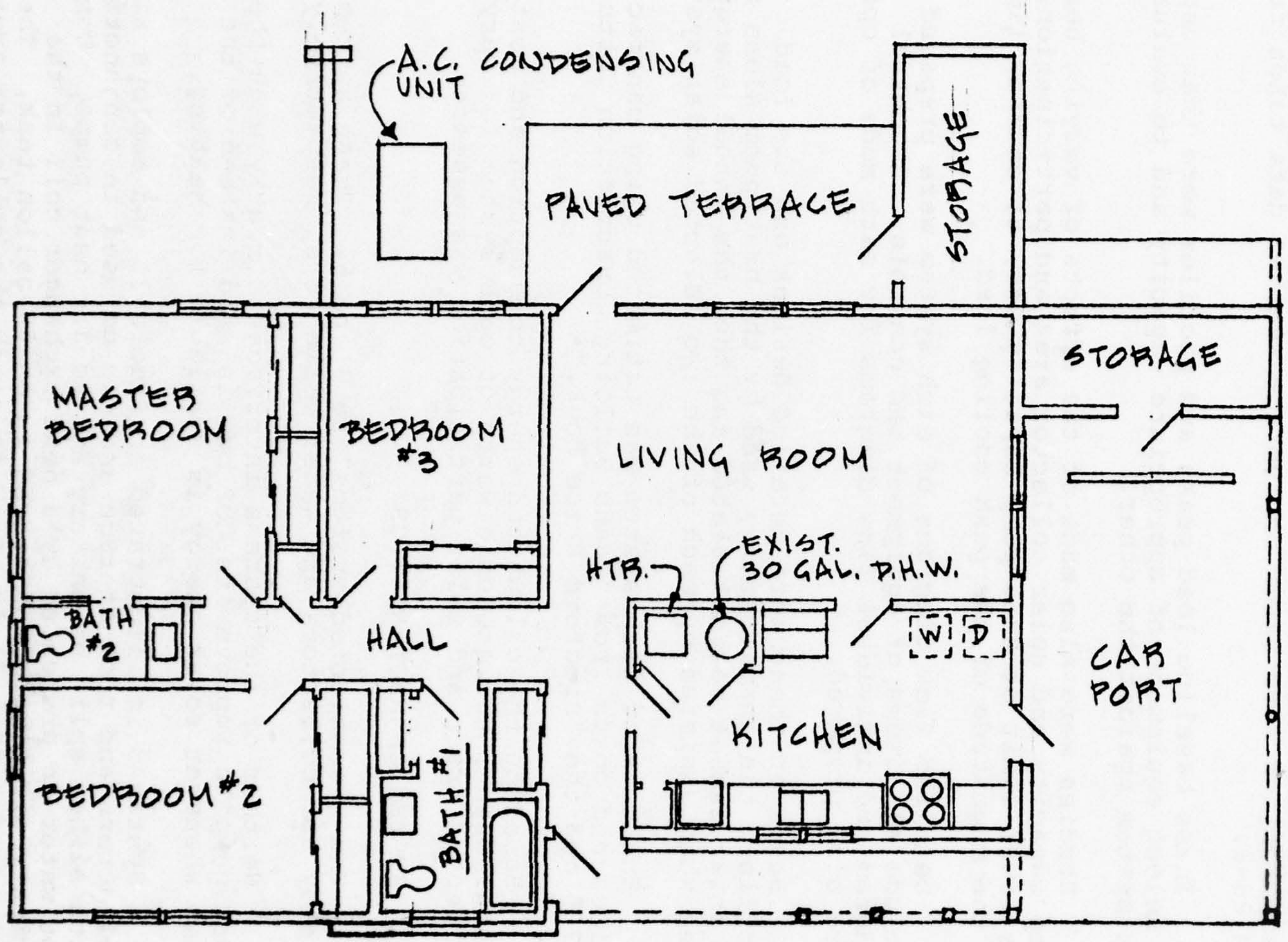


Figure 30. Typical Single Family House Floor Plan

Peak heating and cooling loads for the typical building were calculated for design conditions in Little Rock, Arkansas.

Yearly load profiles for heating, cooling and domestic hot water were constructed. Calculations to establish the load profiles were based on average weather data taken from AFM 88-8.

These baseline load peaks and profiles were then used to select equipment of appropriate capacity and to evaluate one system against the other.

Studies were also made of the effects of varying heat pump capacity and solar collector area and participation. The lower limit of heat pump capacity was, of course, limited by the magnitude of the peak cooling load.

Detailed flow diagrams of each system were prepared to include all items of equipment and controls. A control diagram and individual flow diagrams for each mode of operation were also prepared.

Based on these more detailed designs and the load profiles, the annual energy used by the heat pump alone with no solar assist was calculated and then the annual energy used when assisted by each of the two selected solar systems.

The cost of each system was estimated using manufacturers prices and costs from "Means Building Construction Costs 1976" and the "Bradford Price Book."

Based on the calculated energy consumption and cost of installation, the economic worth of each system for varying collector areas and solar participation was assessed.

3.4 SYSTEM CHARACTERISTICS

The two selected systems are 3 and 6. These systems employ air collectors and water collectors, respectively.

Neither of the systems interfaces directly with the heat pump to improve its COP but is used instead of the heat pump whenever solar energy is available for heating.

System 3 is illustrated on Figure 31 and employs air collectors and rock storage and can be used in conjunction with either split or unitary air to air heat pumps. Domestic hot water is preheated by a heat exchanger coil in the primary air stream and stored in an 80-gallon tank, The primary fan (Fan 1) is sized to match the collector area and its capacity will vary directly as the number of panels

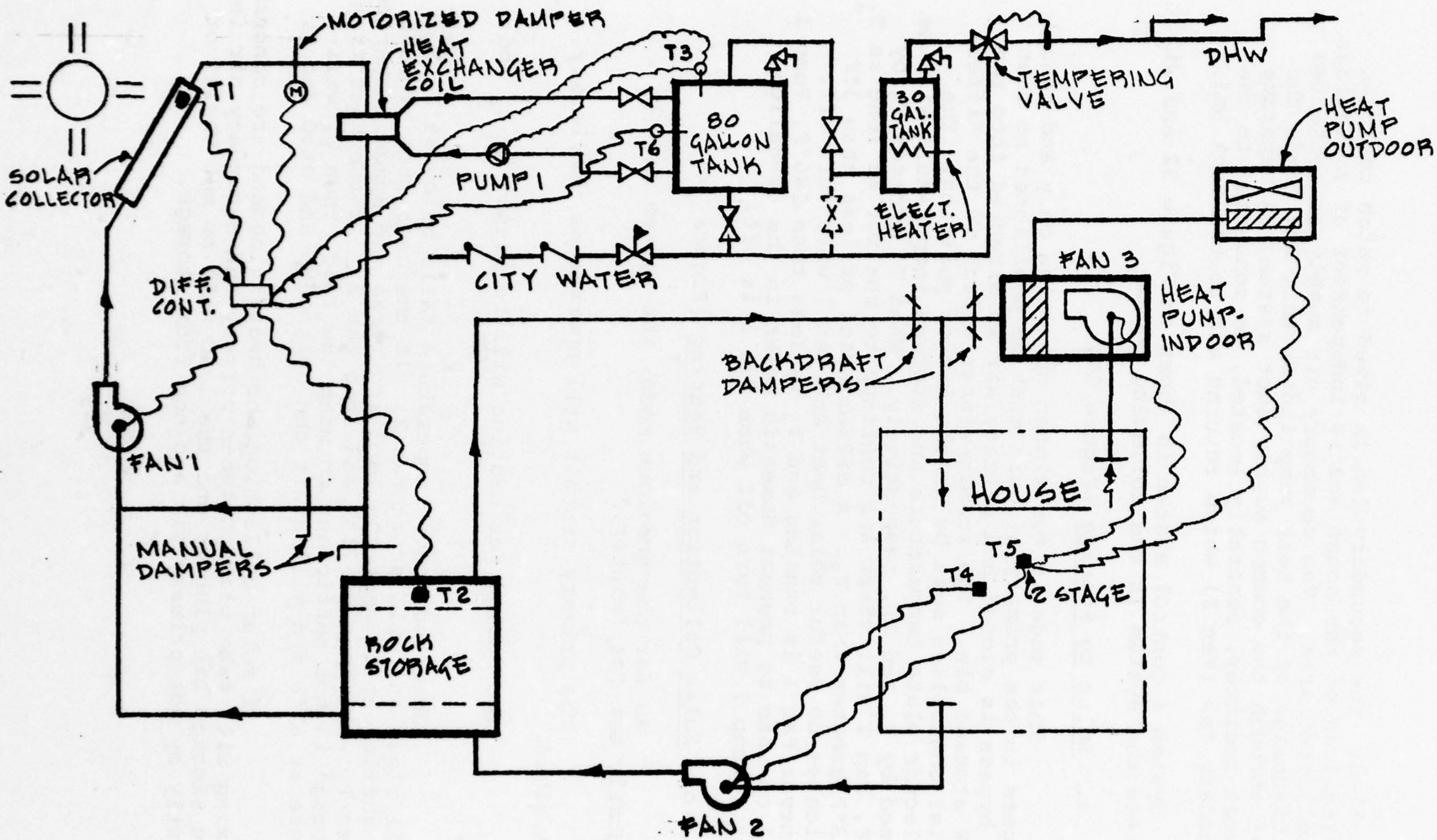


Figure 31. System No. 3 Flow Diagram

installed. The secondary fan is sized to match the peak heating load of the house and is independent of variations in collector area. The secondary air supply is connected to the discharge of the heat pump indoor unit and feeds the house through the common supply duct system. A separate exhaust register, centrally located, is connected to the secondary fan (Fan 2) which returns air to the rock bed.

System 3 control system is shown on Figure 32 and will operate the system in the following modes:

a. Solar to Storage (Figure 33)

This mode is for winter operation only and manual dampers in the primary circuit must be positioned so that the bypass is closed and primary air is directed into the rock storage bin. The summer/winter switch in the differential controller must be in the winter position. The collector plate temperature and storage temperature will be sensed by T_1 and T_2, respectively. When T_1 exceeds T_2 by 20°F, Fan 1 will start and continue to run for as long as T_1 is 3°F greater than T_2. A differential of less than 3°F indicates no useful solar heat and Fan 1 will turn off. Whenever Fan 1 is running and T_3 is less than 180°F, Pump 1 will operate to preheat domestic water in the 80-gallon tank. Pump 1 will turn off when Fan 1 is off.

b. Solar Collection and Heating (Figure 34).

As for the previous mode, the system must be manually set for "winter."

The primary circuit will operate as previously described.

The demand for heating will be sensed by T_4 and T_5.

When the space temperature falls below 69°F, T_4 will close and will start Fan 2. If the heat from storage is sufficient to meet the load and raise the space temperature, then T_4 will open at 71°F and stop Fan 2. If the heat from storage is not sufficient to meet the load, then T_5 will close at 67°F and will start the heat pump and stop Fan 2.

If solar collection and heating demand are concurrent, mixing will take place between primary and secondary air in the storage bin plenums and the load will be met wholly or partly by hot primary air and not from storage.

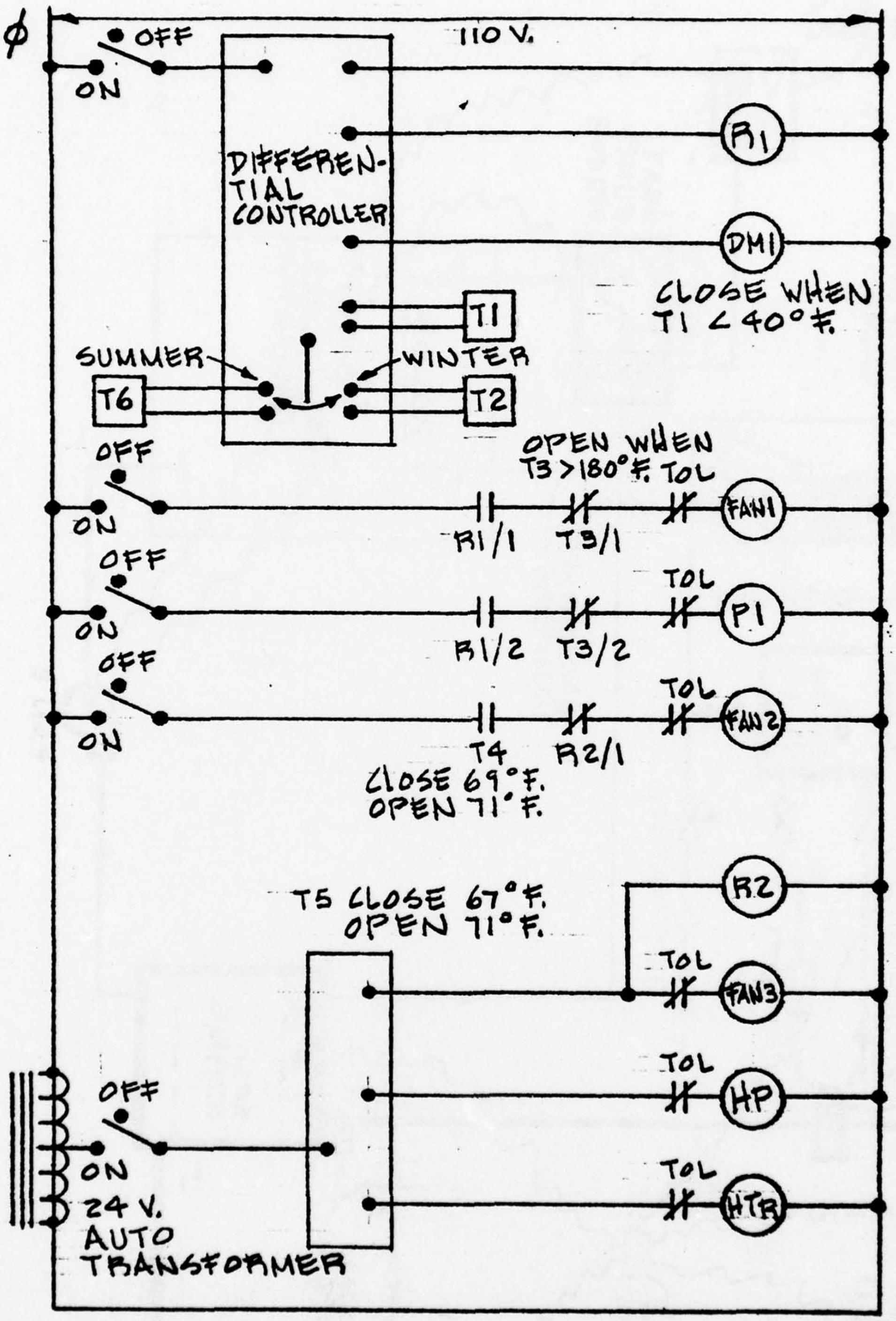


Figure 32. System No. 3 Control Diagram

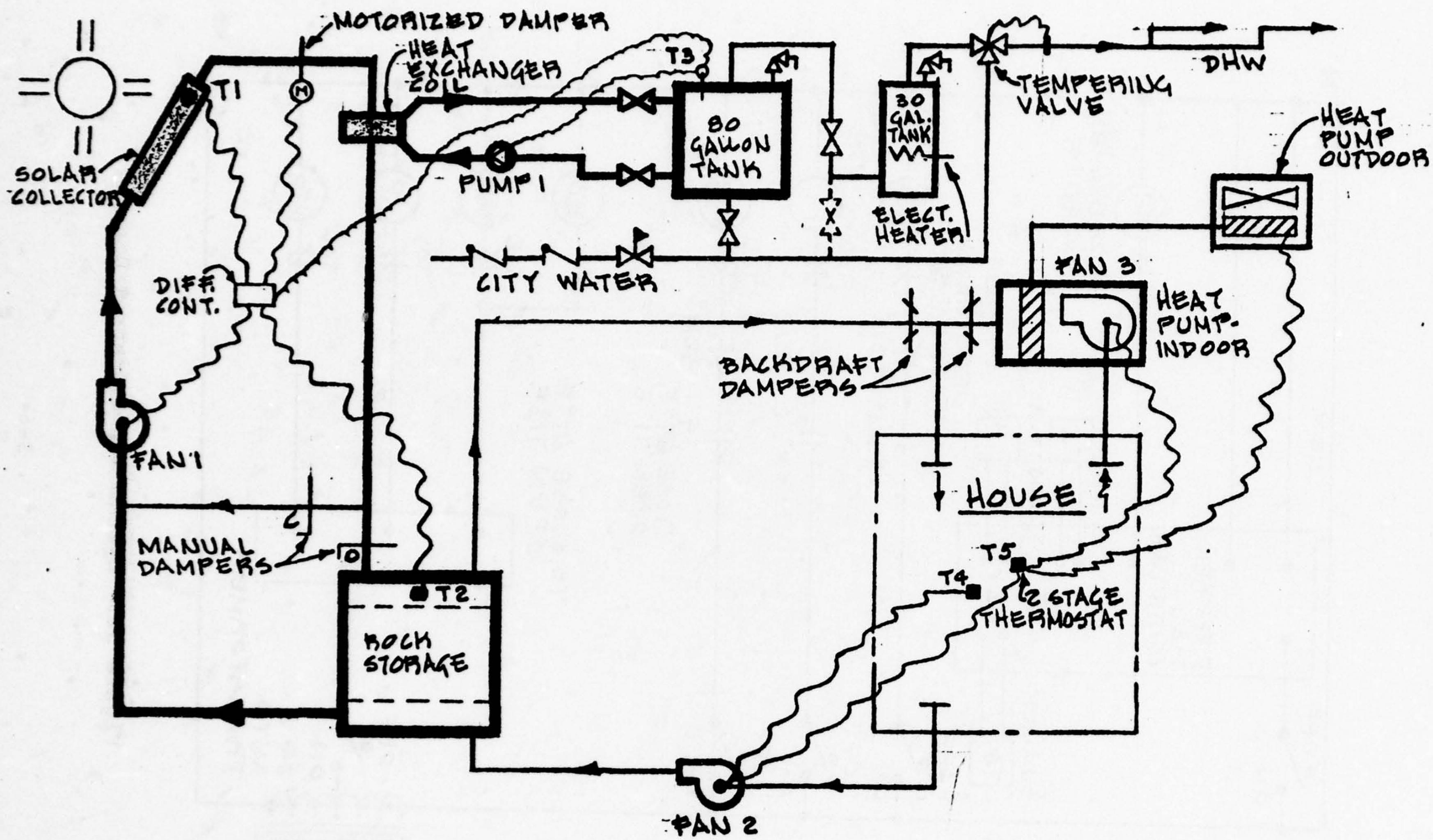


Figure 33. System No. 3 Solar to Storage Mode (Winter Only)

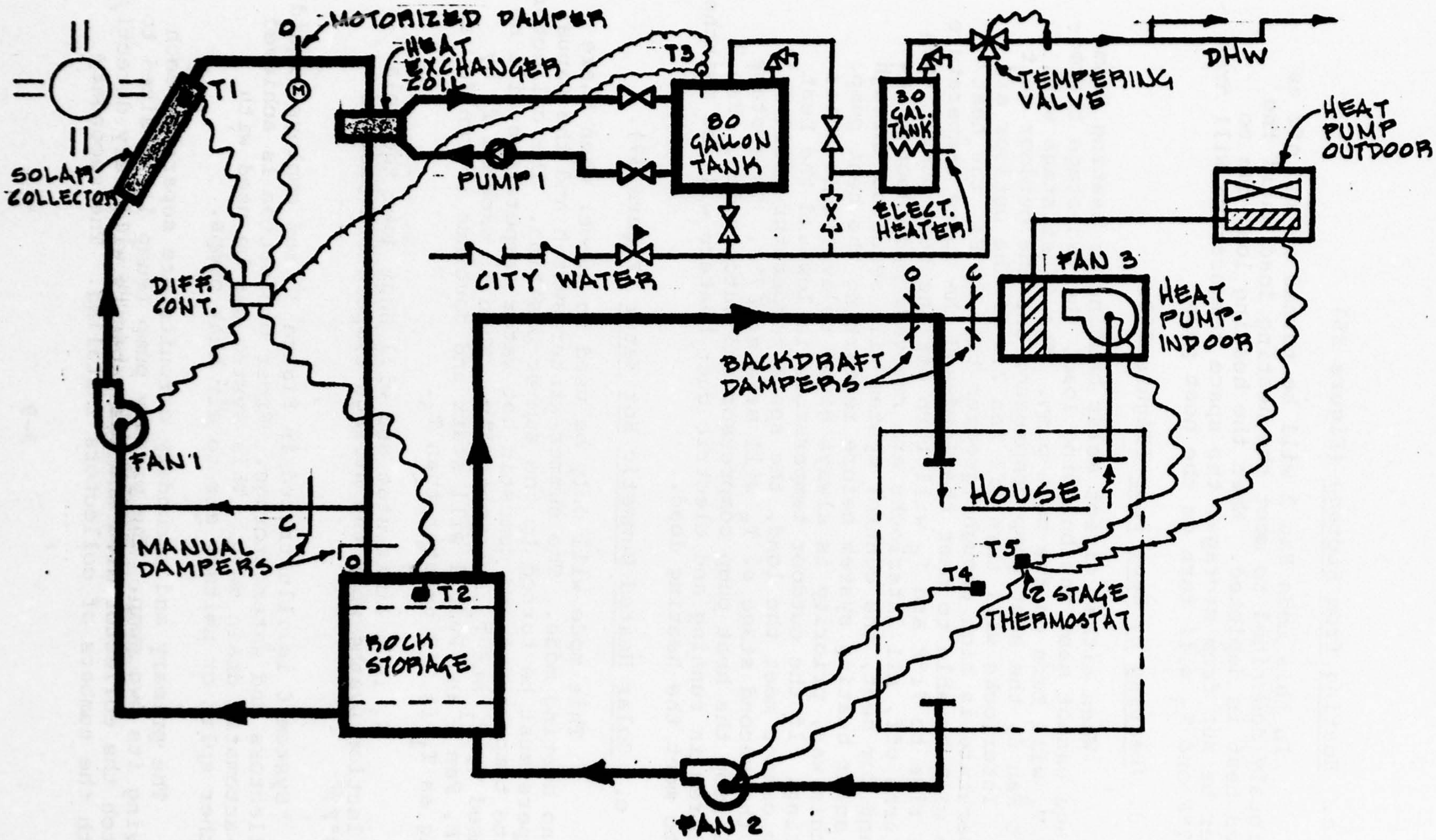


Figure 34. Systen No. 3 Solar Collection and Heating Mode

c. Heating from Storage (Figure 35)

In this mode Fan 2 will be turned on and off as previously described to meet the heating load until the stored heat is depleted. When the heating load can no longer be met from storage, the space temperature will drop to 67°F and T_5 will turn on the heat pump.

d. Heating by Heat Pump (Figure 36).

When either direct solar heating or heating from storage cannot meet the building load, the two-stage thermostat T_5 will turn on the heat pump. The first stage will start Fan 3, the heat pump compressor and the outdoor unit fan. Interlocks will turn off Fan 2. If the outdoor air temperature is high enough (greater than 20°F), the heat pump will be able to meet the load and the space temperature will rise to 71°F and T_5 will turn off the heat pump. When T_5 turns off, all interlocks are removed, and, upon a new demand for heat, the control system will sequence through the solar heating system before restarting the heat pump. In this way, priority is always given to available solar heating. If the outdoor temperature is low and the heat pump cannot meet the load, the space temperature will fall and the second stage of T_5 will make at 67°F. This stage will stop the heat pump compressor and outdoor fan. Fan 3 will remain running and electric duct heaters will be switched on to meet the heating load.

e. Solar Heated Domestic Hot Water (Figure 37).

This mode will only be used in summer when there is no heating mode. The summer/winter switch and the manual dampers must be turned to the summer position. The collector plate temperature and domestic hot water temperature will be sensed by T_1 and T_6, respectively. When T_1 exceeds T_6 by 20°F, Fan 1 and Pump 1 will start and continue to run for as long as T_1 is 3°F greater than T_6.

T_3 is a limit stat and will shut down the solar collection system when the storage temperature exceeds 180°F.

System 6 is illustrated in Figure 38 and employs liquid collectors and water storage. Frost protection is achieved by automatic drain down. This system can be used with either split or unitary air to air heat pumps.

The primary and secondary circuits are separate, each having its own pump. The primary pump (Pump 1) is sized to match the collector area and its capacity will vary directly with the numbers of collectors installed. The secondary

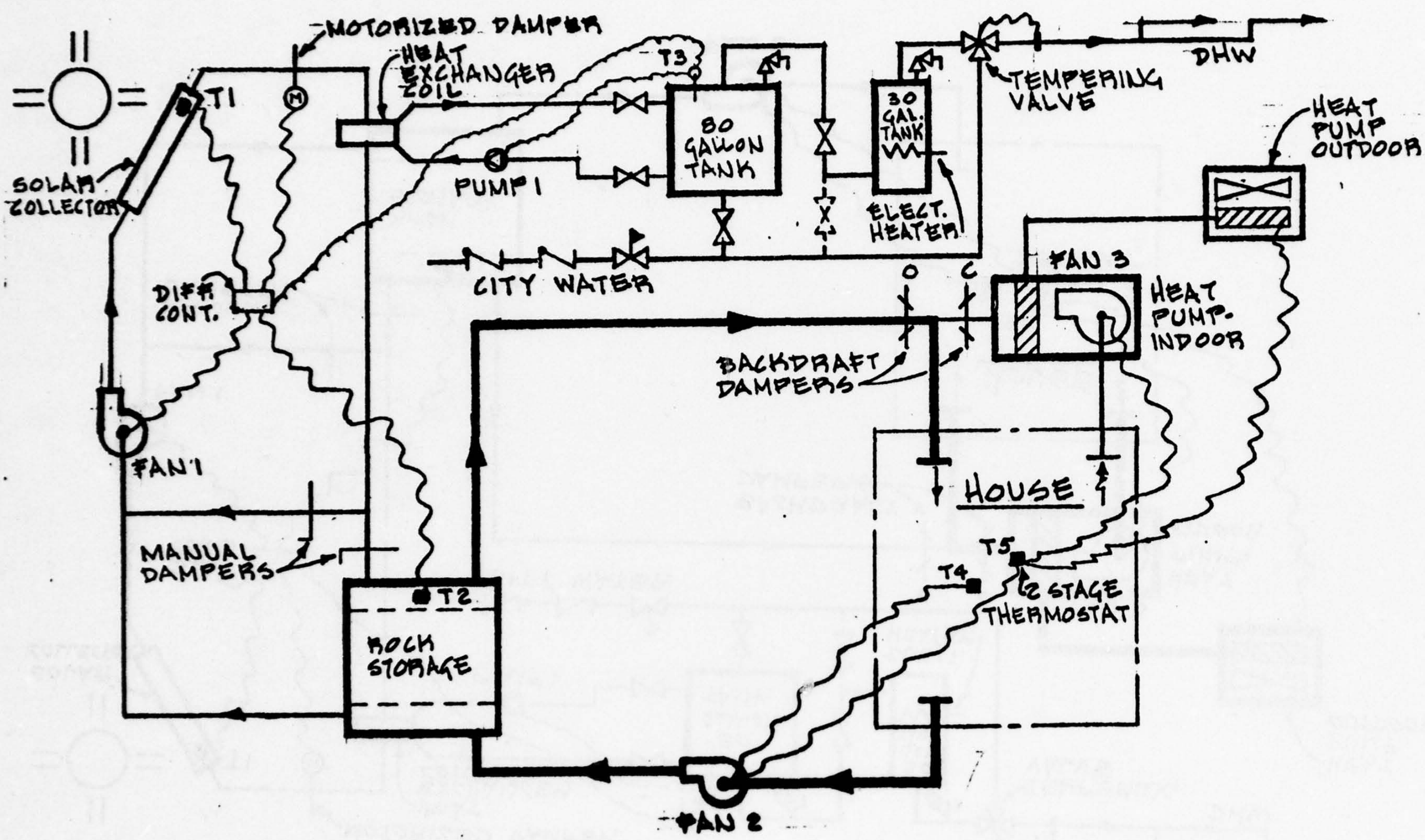


Figure 35. System No. 3 Heating From Storage

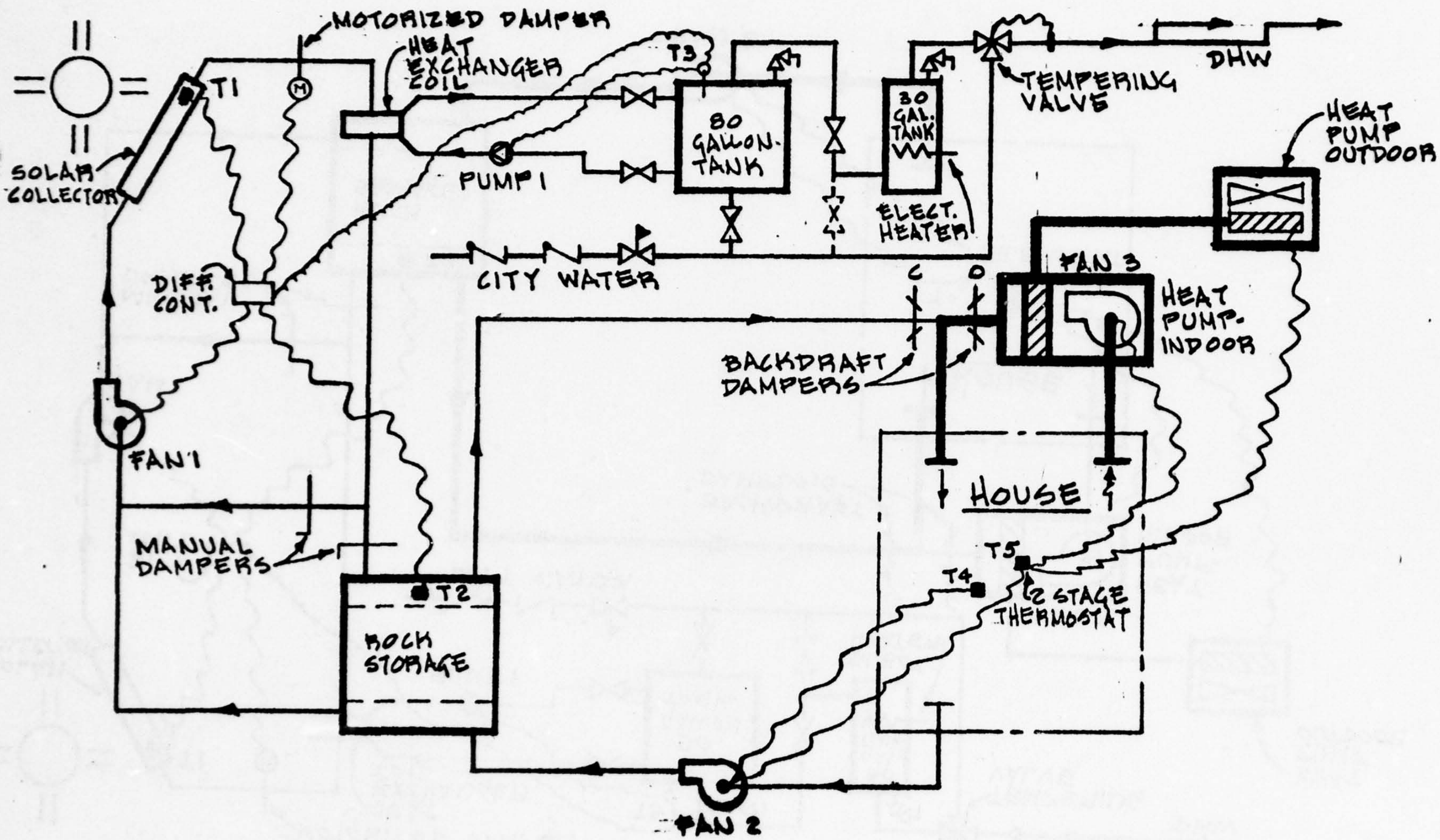


Figure 36. System No. 3 Heating by Heat Pump

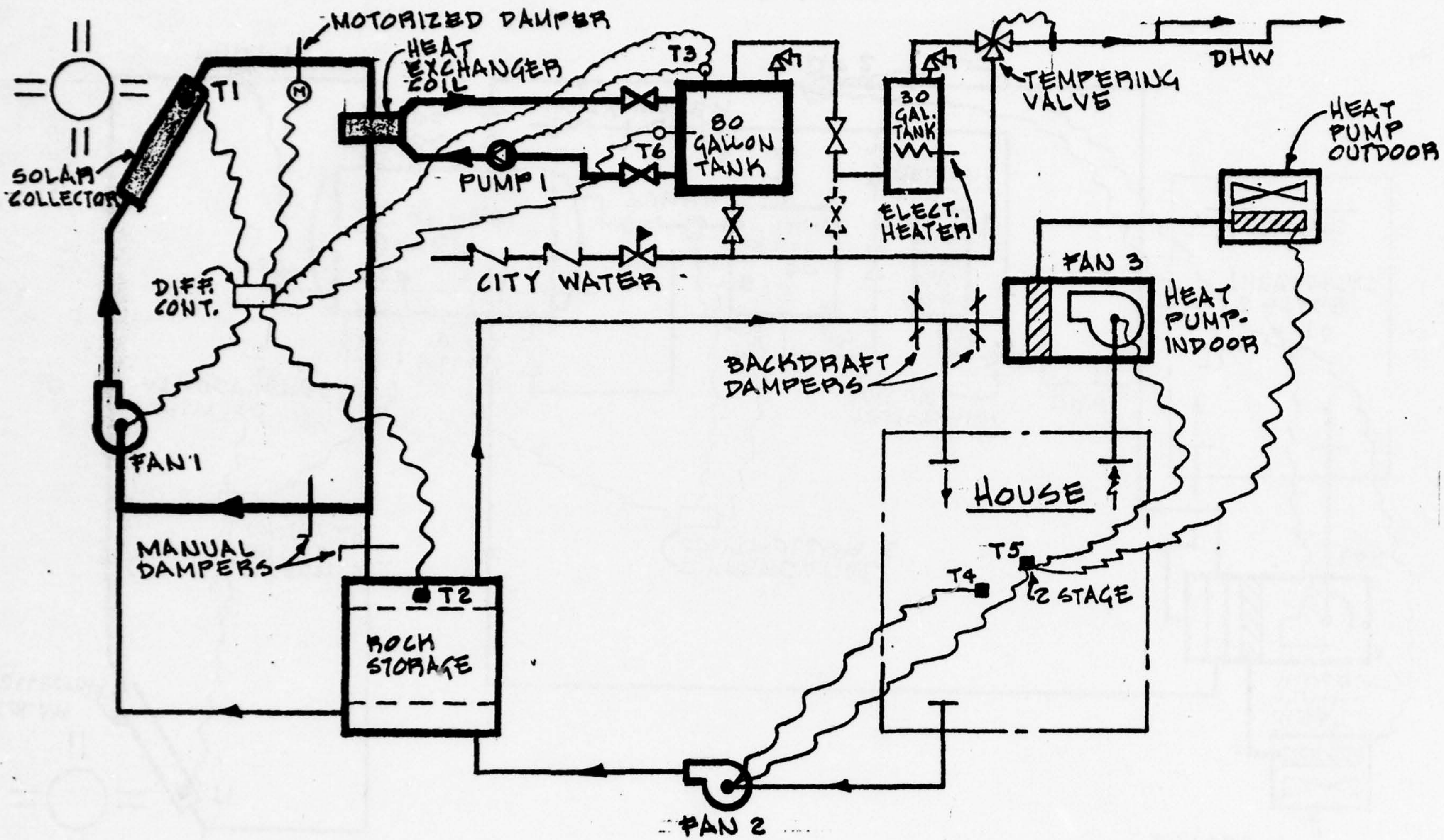


Figure 37. System No. 3 Solar Domestic Hot Water (Summer Only)

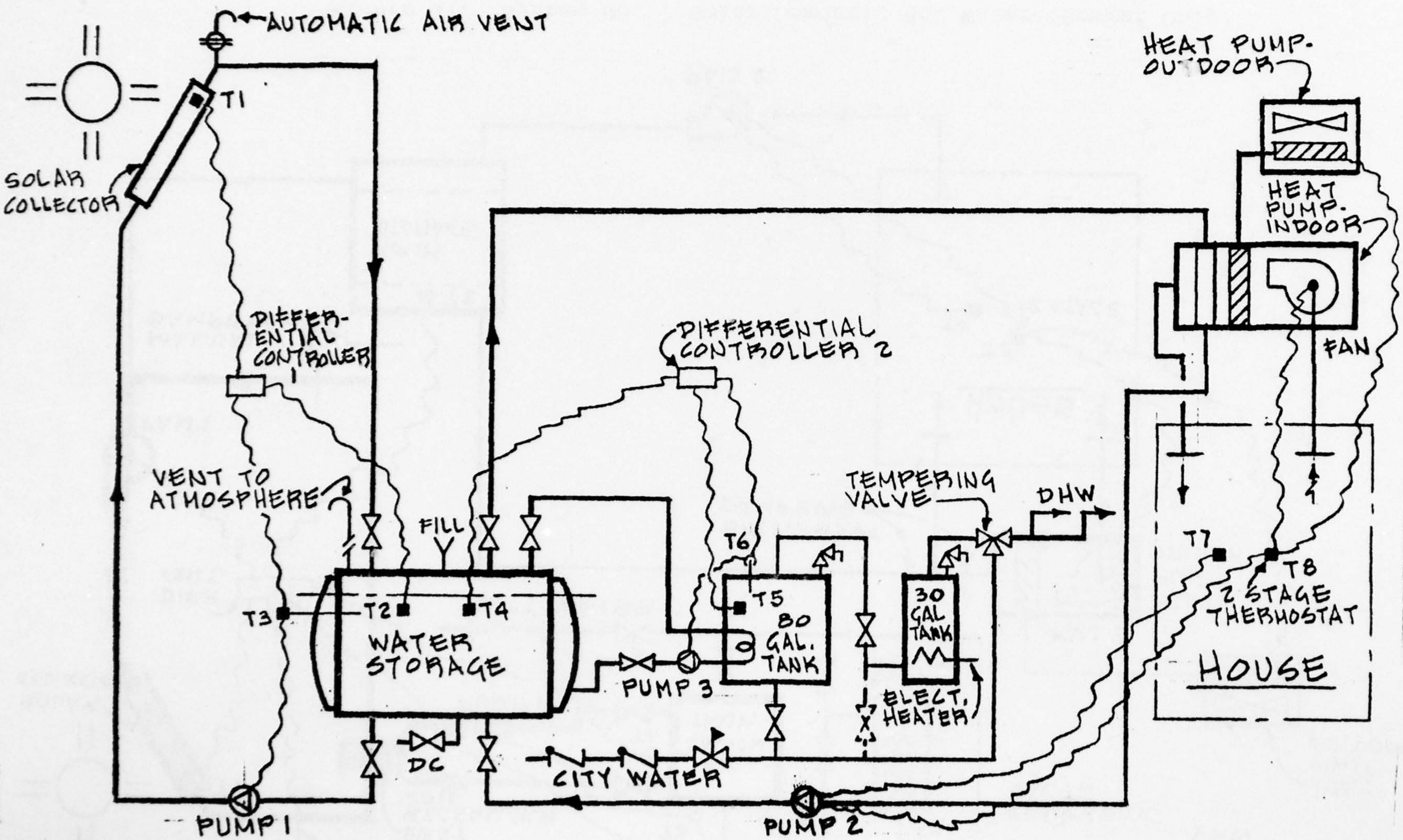


Figure 38. System No. 6 Flow Diagram

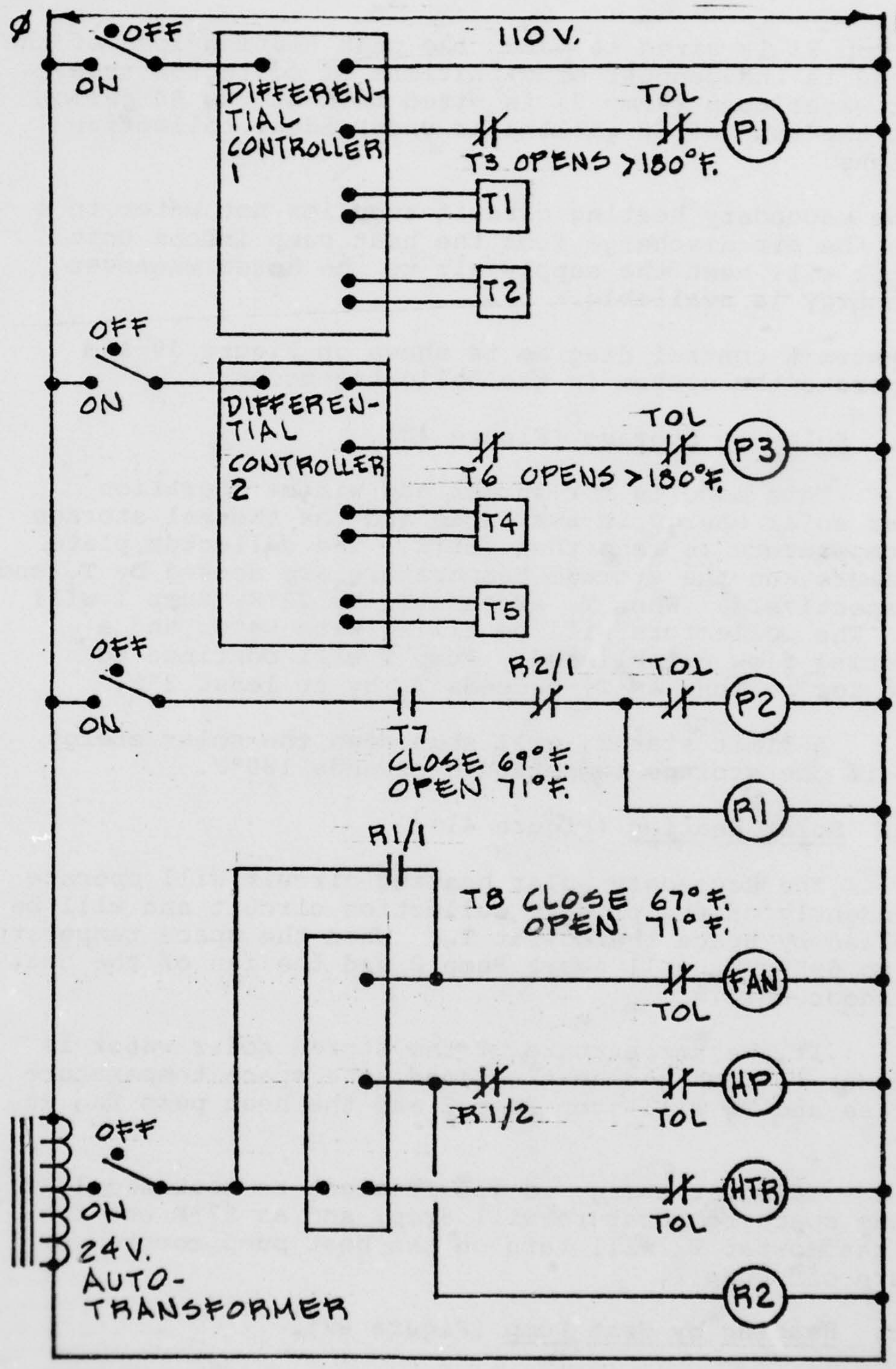


Figure 39. System No. 6 Control Diagram

pump (Pump 2) is sized to match the peak heating load of the house and is independent of variations of collector area. The hot water pump (Pump 3) is sized to heat the 80 gallon tank at the rate of 20 gallons/hr under ideal collection conditions.

The secondary heating circuit supplies hot water to a coil in the air discharge from the heat pump indoor unit. This coil will heat the supply air to the house whenever solar energy is available.

System 6 control diagram is shown on Figure 39 and will operate the system in the following modes:

a. <u>Solar to Storage</u> (Figure 40).

This mode is for summer and winter operation whenever solar energy is available and the thermal storage tank temperature is less than 180°F. The collector plate temperature and the storage temperature are sensed by T_1 and T_2, respectively. When T_1 exceeds T_2 by 20°F, Pump 1 will start. The collectors will be filled with water and a circulating flow established. Pump 1 will continue to operate for as long as T_1 exceeds T_2 by at least 3°F.

A limit stat T_3 will shut down the solar energy system if the storage temperature exceeds 180°F.

b. <u>Solar Heating</u> (Figure 41)

The secondary solar heating circuit will operate independently of the primary collection circuit and will be controlled by space thermostat T_7. When the space temperature falls to 69°F, T_7 will start Pump 2 and the fan of the heat pump indoor unit.

If the temperature of the stored solar water is sufficient to meet the heating load, the space temperature will rise and T_7 will stop Pump 2 and the heat pump fan at 71°F.

If solar energy is insufficient to meet the load, then the space temperature will drop, and at 67°F the two-stage thermostat T_8 will turn on the heat pump compressor and turn off Pump 2.

c. <u>Heating by Heat Pump</u> (Figure 42).

When there is insufficient stored solar energy to meet the heating load, then the two-stage thermostat T_8 will close at 67°F and turn on the heat pump compressor and both of the inside and outside unit fans.

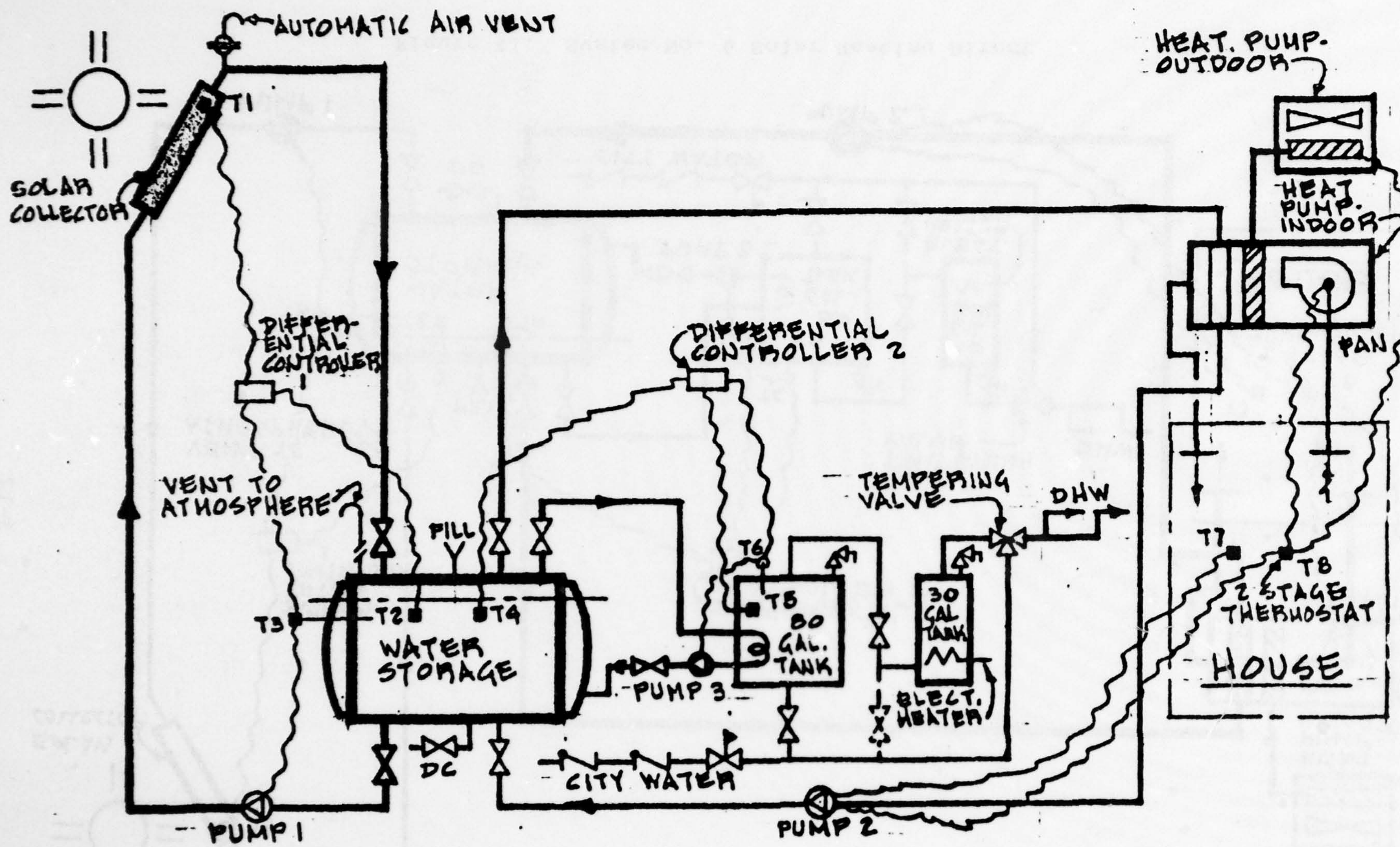


Figure 40. System No. 6 Solar to Storage Mode

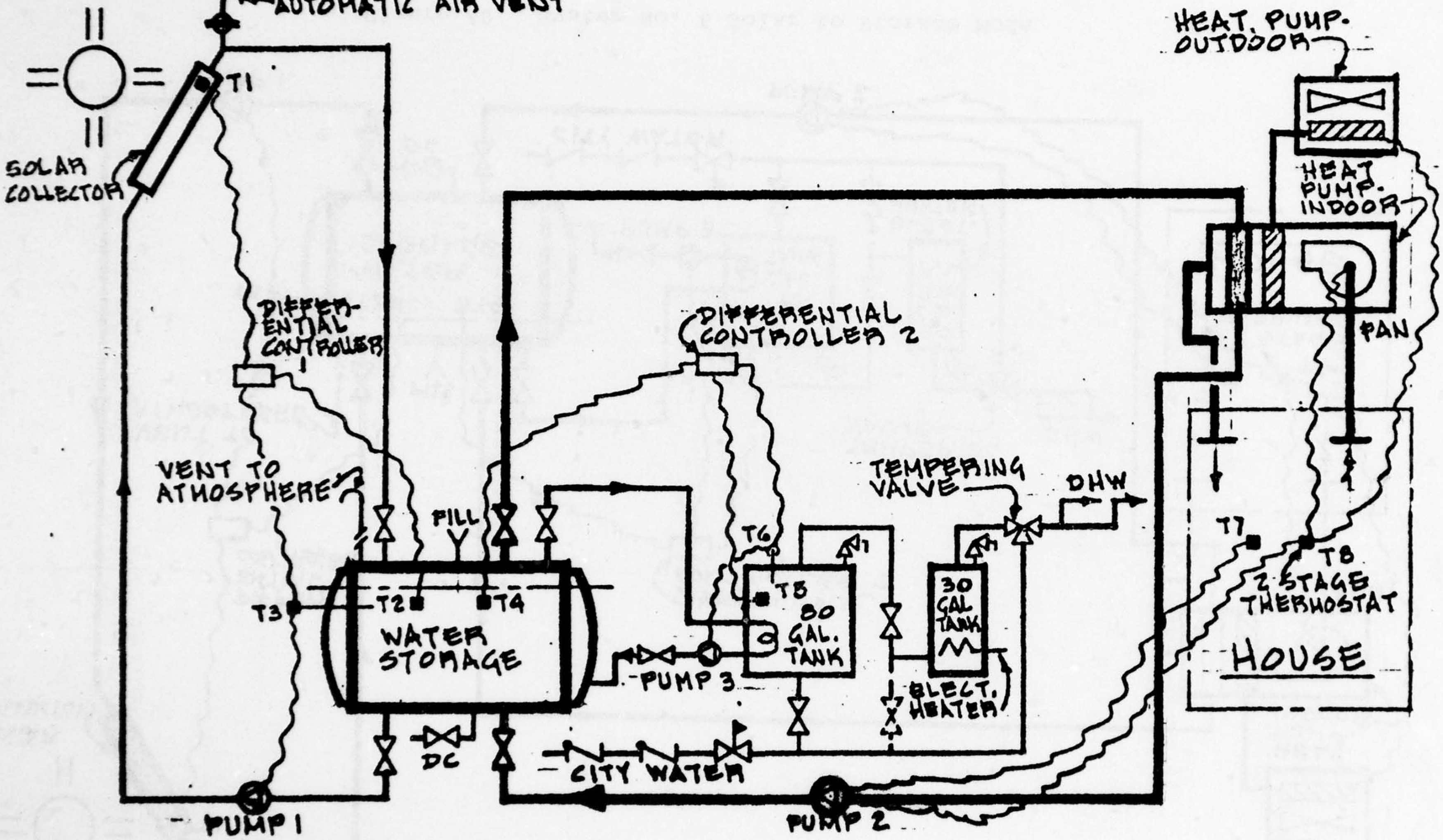


Figure 41. System No. 6 Solar Heating Direct

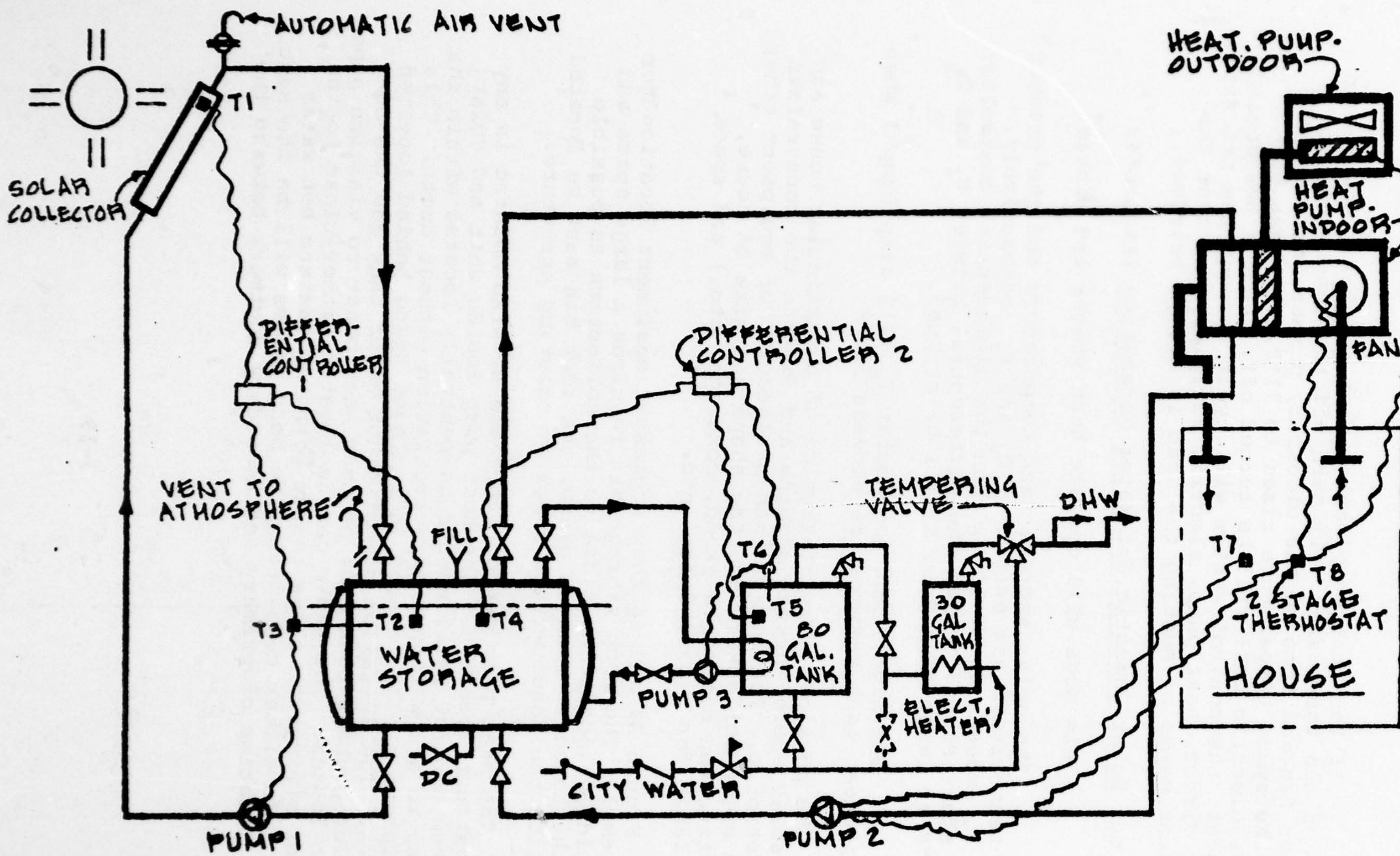


Figure 42. System No. 6 Heating by Heat Pump

If the heat pump cannot meet the load, the second stage of the thermostat will turn off the compressor and outdoor fan and turn on the electric, direct heating coil. When the space temperature rises to 71°F, the heat pump and/or electric coil will be turned off. Upon a new demand for heat, the control system will cycle through the priority--solar direct, heat pump, electric auxiliary--so that the cheapest means of meeting the load is always selected.

d. Solar Domestic Hot Water Generation (Figure 43)

This mode will operate both summer and winter.

The solar storage tank temperature and the preheat tank temperature are sensed by T_4 and T_5, respectively. When T_4 exceeds T_5 by 10°F, then the differential controller will start Pump 3. When the differential between T_4 and T_5 is 3°F or less, then Pump 3 will be stopped.

The high limit thermostat T_6 will stop Pump 3 when the preheat tank temperature exceeds 180°F.

The physical characteristics of a particular house and the solar energy system combine and dictate the constraints of retrofit. The constraints on location of equipment other than solar collectors are the shape and size of house, construction materials (wood, concrete, etc.) and space available for pipes, ducts, etc.

Figure 44 and 45 show suggested equipment location for System 3. The rock storage bin requires a large space and should be located as close to the collectors as possible. If the house is slab on grade, the rock bin must be located outside the house within a new or existing structure.

The heat pump outdoor unit can be slab mounted in any convenient location. The heat pump inside unit and solar system control panel should be centrally located within the house to allow return air circulation without ducts. This equipment can be located in a closet space behind louvered doors. The 30-gallon domestic hot water tank and the 80-gallon preheat tank must be near each other to minimize pipe heat losses. The tanks can be located either close to the solar storage or close to the point of maximum hot water use. In either case, the pipe heat losses will be the same. The location of primary and secondary ductwork between the

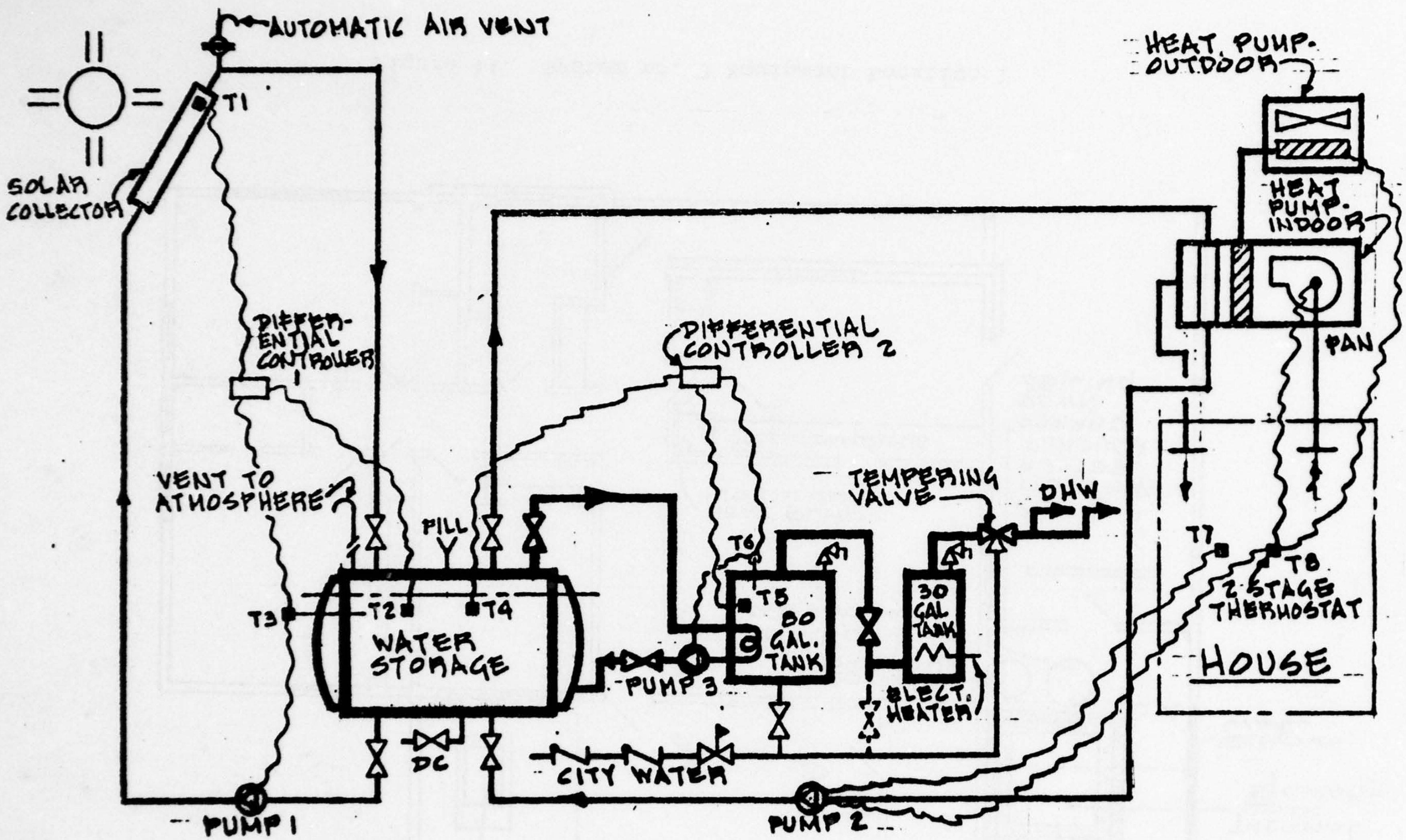


Figure 43. System No. 6 Solar Domestic Hot Water Generation

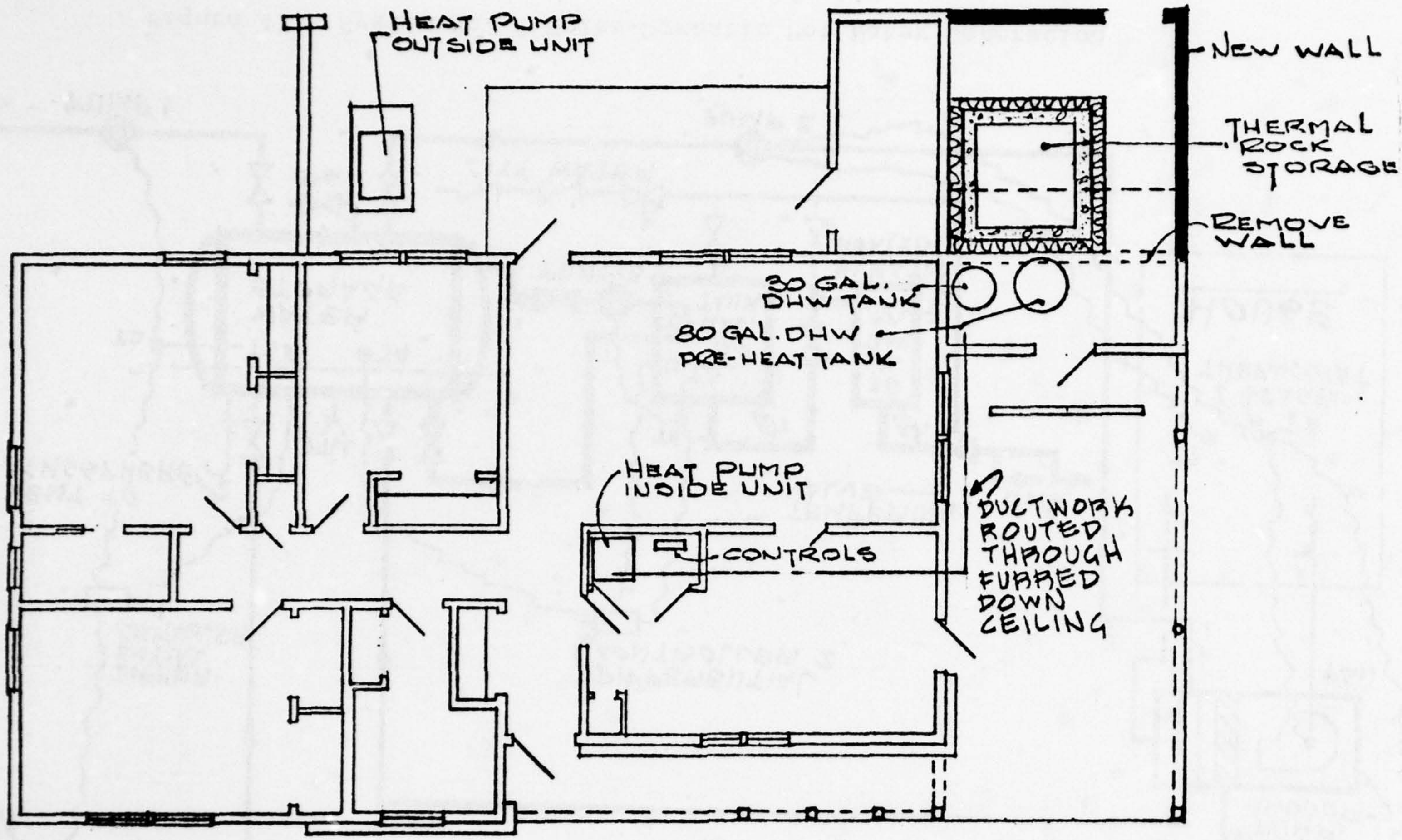


Figure 44. System no. 3 Equipment Location 1

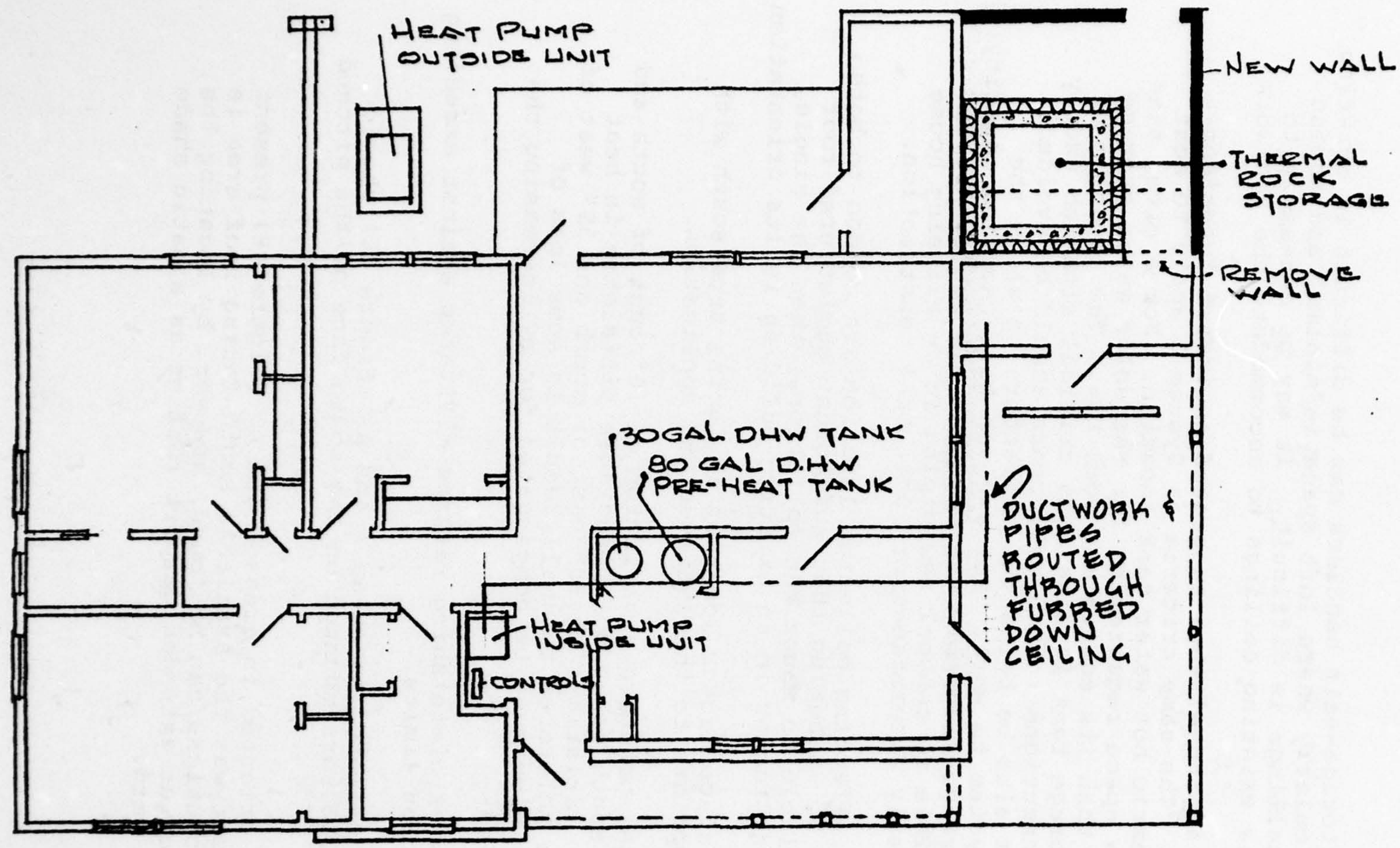


Figure 45. System No. 3 Equipment Location 2

collectors--storage--air handlers can be difficult in existing houses, particularly where loft space is minimal and access through the ceilings is difficult. It may be necessary to fir down below existing ceilings to accommodate the ductwork.

Figures 46 and 47 show suggested equipment locations for System 6. The same criteria as System 3 apply to heat pump and domestic hot water tank location. For a given size of system the space requirement for the solar water storage tank is less than its equivalent rock bin. The location of the water storage tank is also less critical although unduly long pipe connections to the collectors should be avoided. The tank must also be below the collectors to allow the drain-down system to work. For a given heat-carrying capacity, pipe sizes are considerably smaller than air ducts. It is usually possible to conceal new piping in an existing house and avoid costly alterations or additional construction.

Factors affecting collector location are common to both System 3 and 6. Assuming that sufficient uncluttered roof area is available to mount the collectors, then the single, most important factor in an existing building is its orientation.

The ideal collector location is facing true south with a tilt appropriate to the latitude and application.

In actual practice a variation of 10° east of south and 15° west of south makes no appreciable difference in heat collection. Variations of 30° east of south and 35° west of south can be tolerated but will result in some loss of performance which must be compensated for by increasing the collector area.

The law of diminishing returns militates against exceeding these variation limits.

Houses oriented in areas A and A_1 (Figure 48) should have the collectors mounted on the south face of the pitched roof.

Houses oriented in areas B and B_1 (Figure 48) present more difficulty as the available south facing roof area is limited. Solutions can be found, however, by locating the collectors on an extended carport roof or as a patio shade in the back yard.

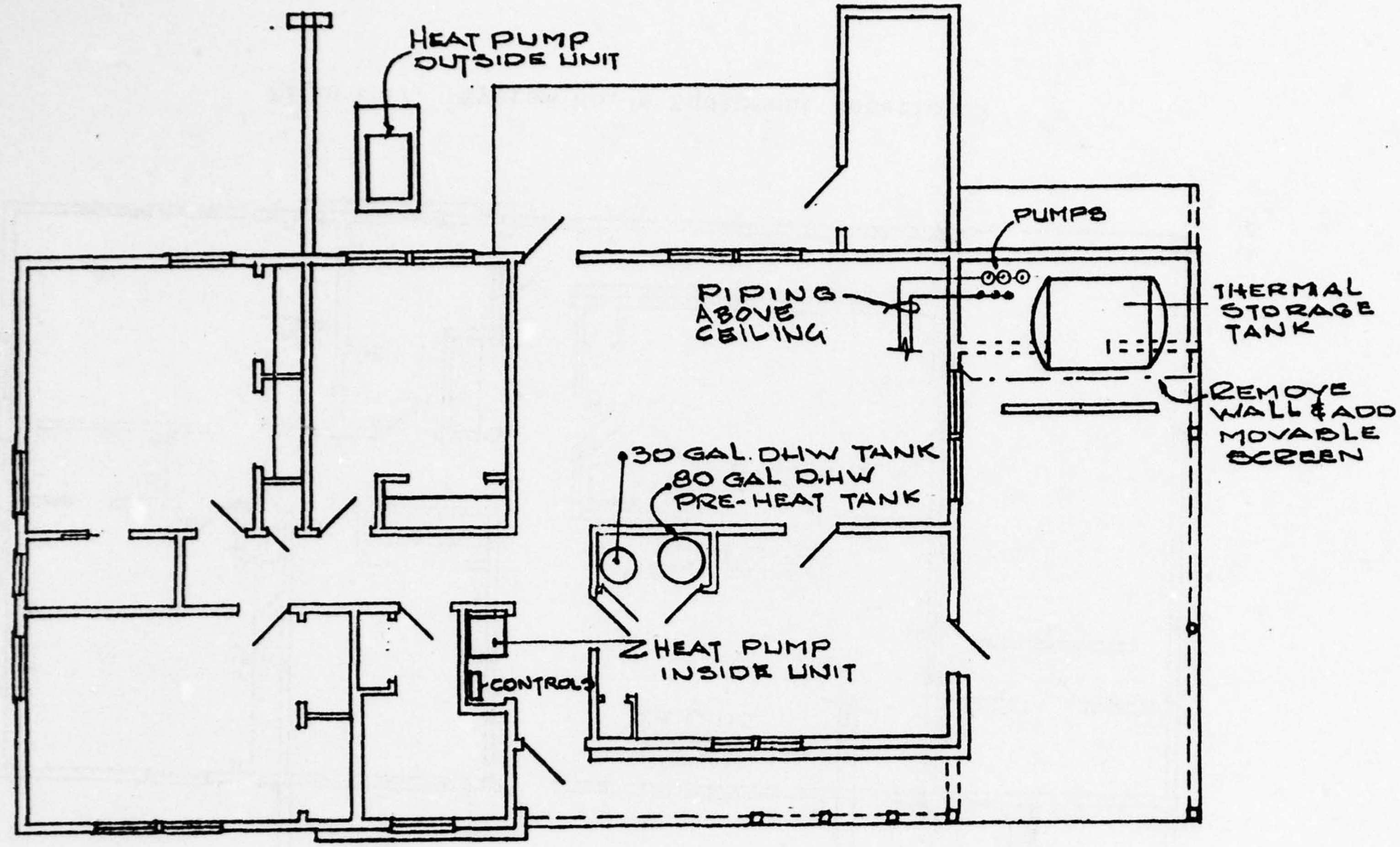


Figure 46. System No. 6 Equipment Location 1

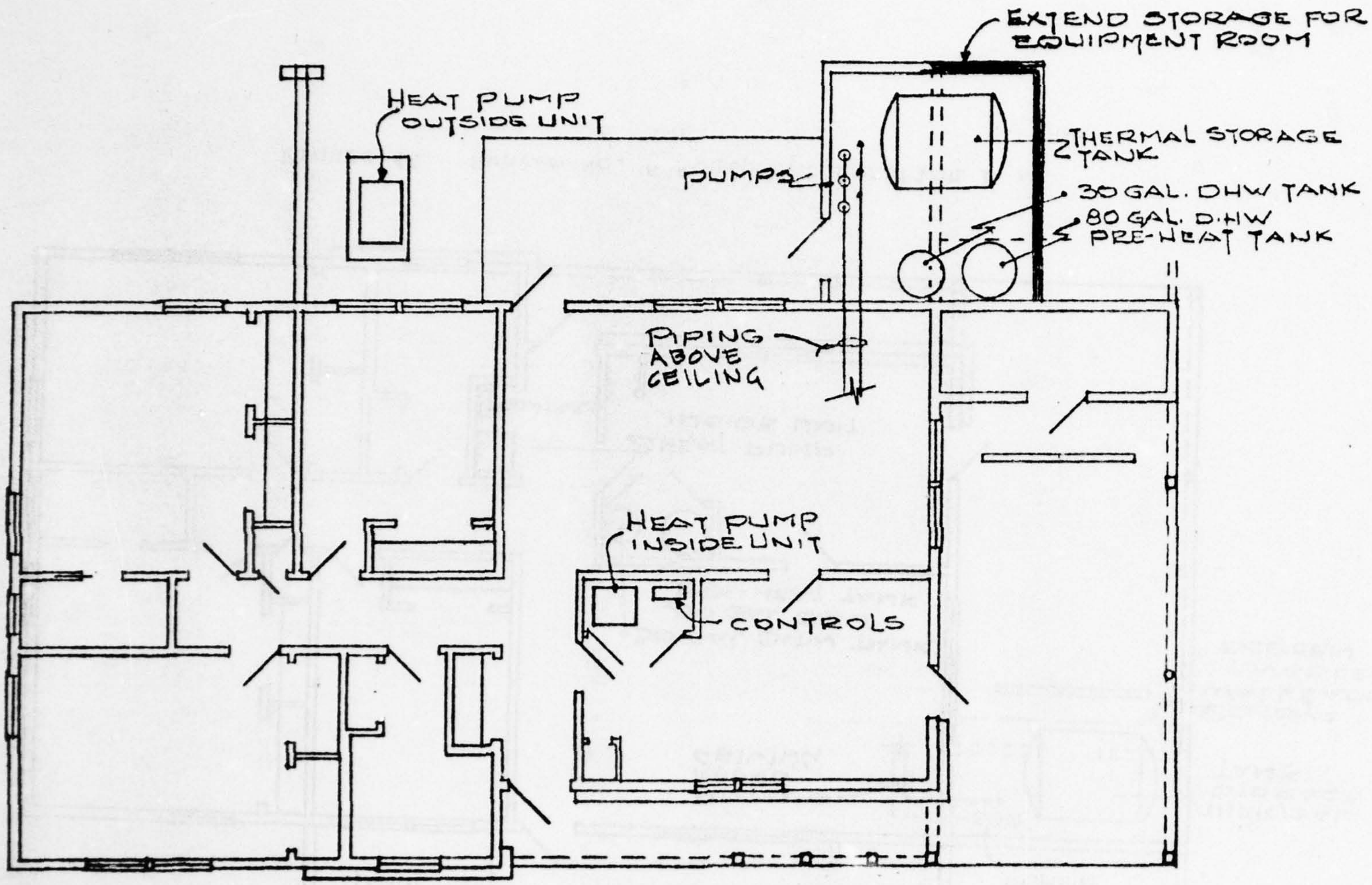


Figure 47. System No. 6 Equipment Location 2

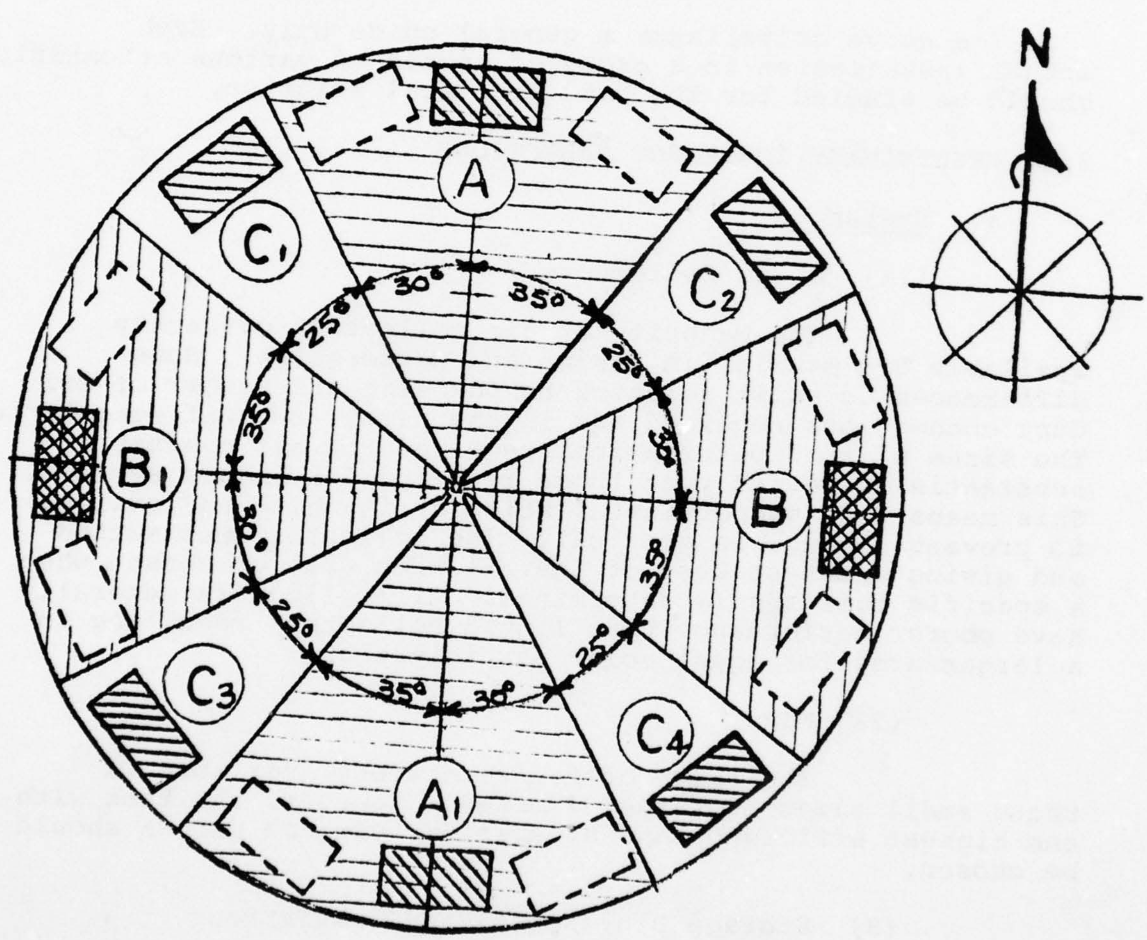


Figure 48. Orientation Chart

Houses oriented in areas C_1, C_2, C_3, and C_4 (Figure 48) have no roof area facing in an acceptable direction.

In these cases the collectors should be mounted as free-standing units or combined with a nearby building of acceptable orientation.

The above criteria are a general guide only. Each actual installation in a group of houses of various orientations should be studied for the best practical solution.

3.5 PRELIMINARY EQUIPMENT SELECTIONS

a. System 3

(1) Solar Collectors

The majority of air collectors presently available are similar in design and performance. Some differences do exist relating to the size and number of air duct connections required and in the number of roof penetrations. The sizes of roof penetrations required for air systems are substantially larger than those required for liquid systems. This means that proper installation of flashing and sealing to prevent leakage is critical. The collector best suited and giving greatest ease of installation will be chosen when a specific building is determined. Air collectors generally have poorer performance than liquid collectors resulting in a larger area for equal solar participation.

(2) Fans

The range of fan selections available in these small sizes is rather limited. However, the fans with the highest efficiency and steeper performance curves should be chosen.

(3) Storage Bin, Rocks, and Insulation

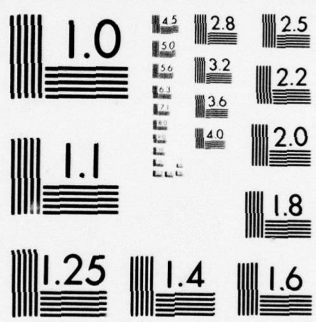
The storage bin can be constructed from wood, poured concrete, or concrete block. It should be insulated on the outside with a minimum of 3 inches of fiberglass. A protective covering should be installed over the insulation. The rocks should be between 3/4 and 1-1/2 inch and thoroughly washed to remove dust before installation. Concrete block construction has been chosen for the system under consideration.

(4) Domestic Hot Water Coil and Tank

The domestic hot water coil should be sized for a recovery rate of 80 gallons in four hours at peak

AD-A058 626 DUBIN-BLOOME ASSOCIATES NEW YORK F/G 13/1
SOLAR ASSISTED HEAT PUMP STUDY FOR HEATING OF MILITARY FACILITI--ETC(U)
JUL 78 F L BEASON, L W STROTHER F08635-76-C-0276
UNCLASSIFIED AFCEC-TR-78-6 NL

2 OF 3
AD
A058626



MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

solar insolation conditions. The 80-gallon preheat tank is a conventional steel shell, glass lined, insulated tank complete with a pressure relief valve and rated to withstand water main pressure.

(5) Controls

The control system consists of a differential controller designed for solar systems and the sensors required for its operation. Several high quality controllers such as the Rho Sigma, are available today. The residence will require two thermostats, one two-stage and one single stage. These must provide accuracy and consistent repeatability as temperature differentials of 2 degrees are used for control system input.

(6) Heat Pump

The heat pump can be either a unitary or split, air-to-air type. A split system is shown in the schematic diagrams. Good performance at low temperatures is essential and is provided by most present-day, state-of-the-art units. The Westinghouse HI-RE-LI and Lennox "Solarmate" heat pumps are typical.

(7) Ducts, Dampers, Fittings, etc.

Ductwork should be oval or round sheet metal, insulated with the equivalent of at least 2 inches of fiberglass. Joints and connections must be sealed tightly. The normally acceptable 10-15 percent leakage rate is excessive for a solar system. Dampers and miscellaneous fittings should be of good quality to provide reliable performance.

b. System 6

(1) Solar Collectors

The selection of the solar collectors should be governed by the design utilization temperature of the system and the temperature range over which the collectors will be operating for the majority of the year. Many high quality collectors are available; however, some distinctions such as piping configuration will be considered during the design phase. Additionally, the KTA tubular collector has the ability to allow for less than optimum roof tilt by rotating the individual tubes. This can be an important feature aesthetically in a retrofit. For additional details, refer to Section II of this report.

(2) Pumps

Pumps will be fractional horsepower, in- line type. The range of selection in these small sizes is limited; however, the highest efficiency pumps with steep performance curves should be chosen.

(3) Storage Tank

The storage tank should be a horizontal, cylindrical steel tank with an interior protective coating and insulated with the equivalent of at least 3 inches of fiberglass. A protective cover should be installed over the insulation.

(4) Domestic Hot Water Tank

See System 3.

(5) Air Handling Unit

See heat pump.

(6) Controls

See System 3. An additional differential controller will be required to operate the domestic hot water system.

(7) Heat Pump

See System 3.

(8) Piping, Valves, Fittings, etc.

Piping should be copper, insulated with the equivalent of at least 1 inch of fiberglass. Valves, air vents and other fittings should be of good quality to give reliable performance.

3.6 SYSTEM CONSTRUCTION COSTS

The system construction costs are based on the tentative equipment selection described in Section 3.5 and on the assumed typical house plan.

Equipment, installation, and construction costs were obtained from manufacturers, Means Construction Costs 1976, and other current cost data sources.

Each system was priced for five sizes ranging from 100 ft^2 to 300 ft^2 collector area.

The breakdown costs and total estimated costs for Systems 3 and 6 are tabulated on Tables 4 and 5, respectively.

3.7 BUILDING LOADS AND HEAT PUMP PERFORMANCE

Building heating and cooling loads were determined on a monthly basis for an 1,150 square foot single family residence in the Little Rock, Arkansas, area. It was assumed that the thermal characteristics of the house would be improved by providing additional wall and roof insulation and double-glazed windows. The domestic hot water load was based on 100 gallons per day heated from 55 to 140°F.

To optimize heat pump selection, peak heating and cooling loads were provided to Lennox Industries, Inc. for computer optimization. The monthly heating loads were modified by the effect of internal and solar gains determined from measured data. The annual heating, cooling, and domestic hot water load distribution is shown in Figure 49. Two heat pump sizes, 25,000 and 29,000 Btu/hr were analyzed and compared on an annual basis. Figures 50 and 51 compare their instantaneous and annual performance, respectively, over a range of outdoor air temperatures. A comparison of energy consumption by the different size heat pumps for the residence under consideration is shown in Table 6. Although the 29,000 Btu/hr unit has a higher instantaneous KWH input over most of the temperature range, this is more than offset by its higher heating and cooling output. The result is that the 29,000 Btu/ hr unit runs for fewer hours to meet the heating and cooling loads. As a result, further analysis incorporating solar energy was based on a system with the 29,000 Btu/hr heat pump installed.

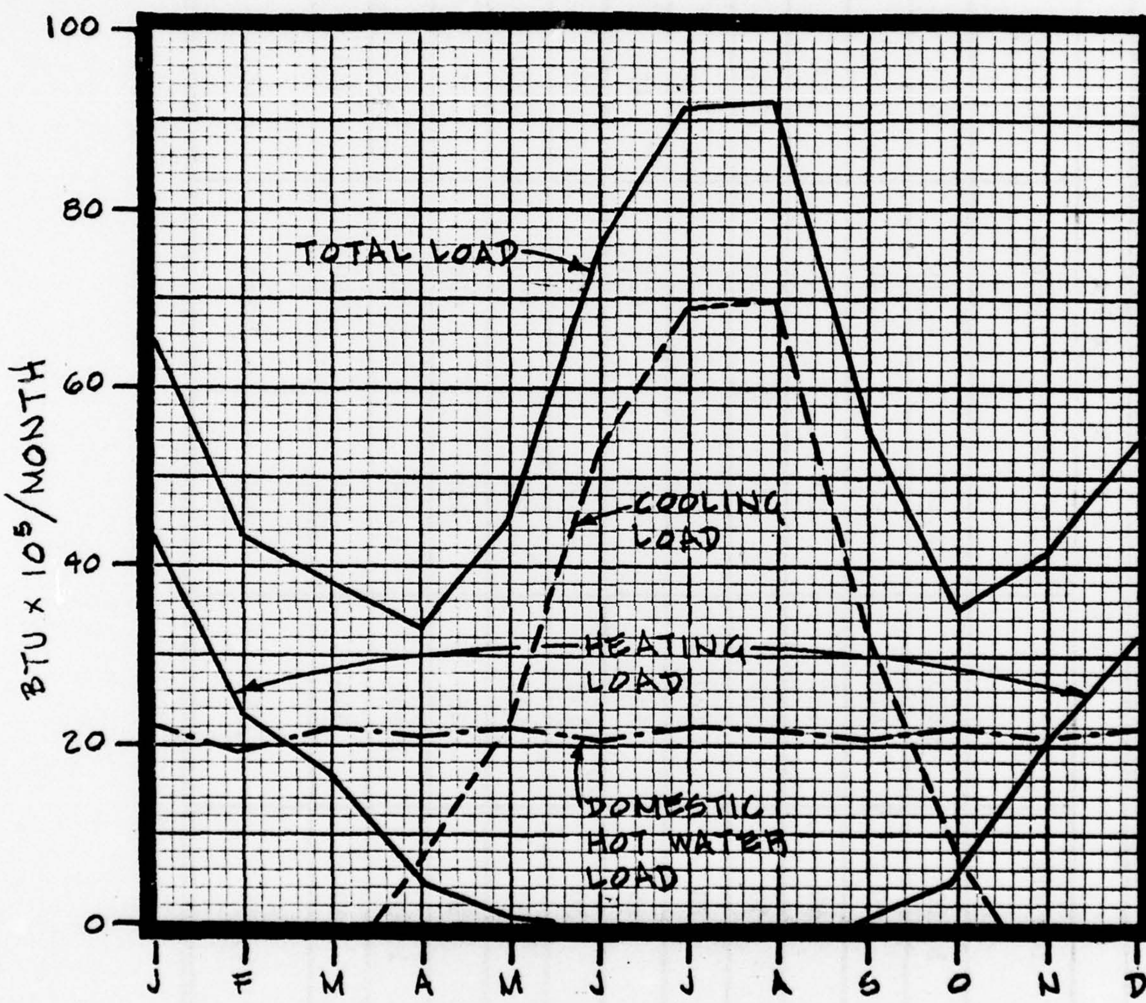
The annual load distribution shown in Figure 49 indicates that the domestic hot water load represents the bulk of the load met by solar energy on a yearly basis. This will be true in geographical areas with climates similar to Little Rock and where energy conservation measures have been taken to reduce the heating load. If no load reduction measures are implemented, this load distribution will occur in more southerly areas. The effect of this type of load distribution on the size of the solar energy system is to make it a function of the magnitude of the domestic hot water load rather than the heating load as is the case in northern areas. This tends to reduce the optimum size of the collector

Table 4. System No. 3 Estimated Cost

ITEM	COLLECTOR AREA				
	100 FT^2	150 FT^2	200 FT^2	250 FT^2	300 FT^2
Collectors	1,420	2,130	2,835	3,550	4,253
Rock Storage	225	340	450	560	680
Primary Fan	230	240	250	270	300
Primary Ducts	500	750	1,000	1,250	1,500
Hot Water Coil	400	400	400	400	400
Hot Water Pump	120	120	120	120	120
Hot Water Tanks	425	425	425	425	425
Secondary Fan	350	350	350	350	350
Secondary Ducts	900	900	900	900	900
Backdraft Dampers	50	50	50	50	50
Heat Pump	1,500	1,500	1,500	1,500	1,500
Controls	190	190	190	190	190
TOTAL	6,310	7,395	8,470	9,565	10,668

Table 5. System No. 6 Estimated Costs

ITEM	COLLECTOR AREA				
	100 FT^2	150 FT^2	200 FT^2	250 FT^2	300 FT^2
Collectors	1,100	1,650	2,200	2,750	3,300
Storage Tank	330	470	625	780	940
Primary Pump	140	150	170	200	230
Primary Pipe	450	500	600	650	700
Hot Water Pump	120	120	120	120	120
Hot Water Tanks	425	425	425	425	425
Secondary Pump	150	150	150	150	150
Secondary Pipe	500	500	500	500	500
Air Coil	450	450	450	450	450
Heat Pump	1,500	1,500	1,500	1,500	1,500
Controls	340	340	340	340	340
<u>TOTAL</u>	5,505	6,255	7,080	7,865	8,655



1,150 SQ. FT. SINGLE FAMILY RESIDENCE
HEATING DEGREE DAYS (BASE 65°F.) 3,300
COOLING DEGREE HOURS (BASE 75°F.) 23,785
PEAK HEATING LOAD 22,140 BTU/HR.
PEAK COOLING LOAD 22,062 BTU/HR.

Figure 49. Annual Load Distribution

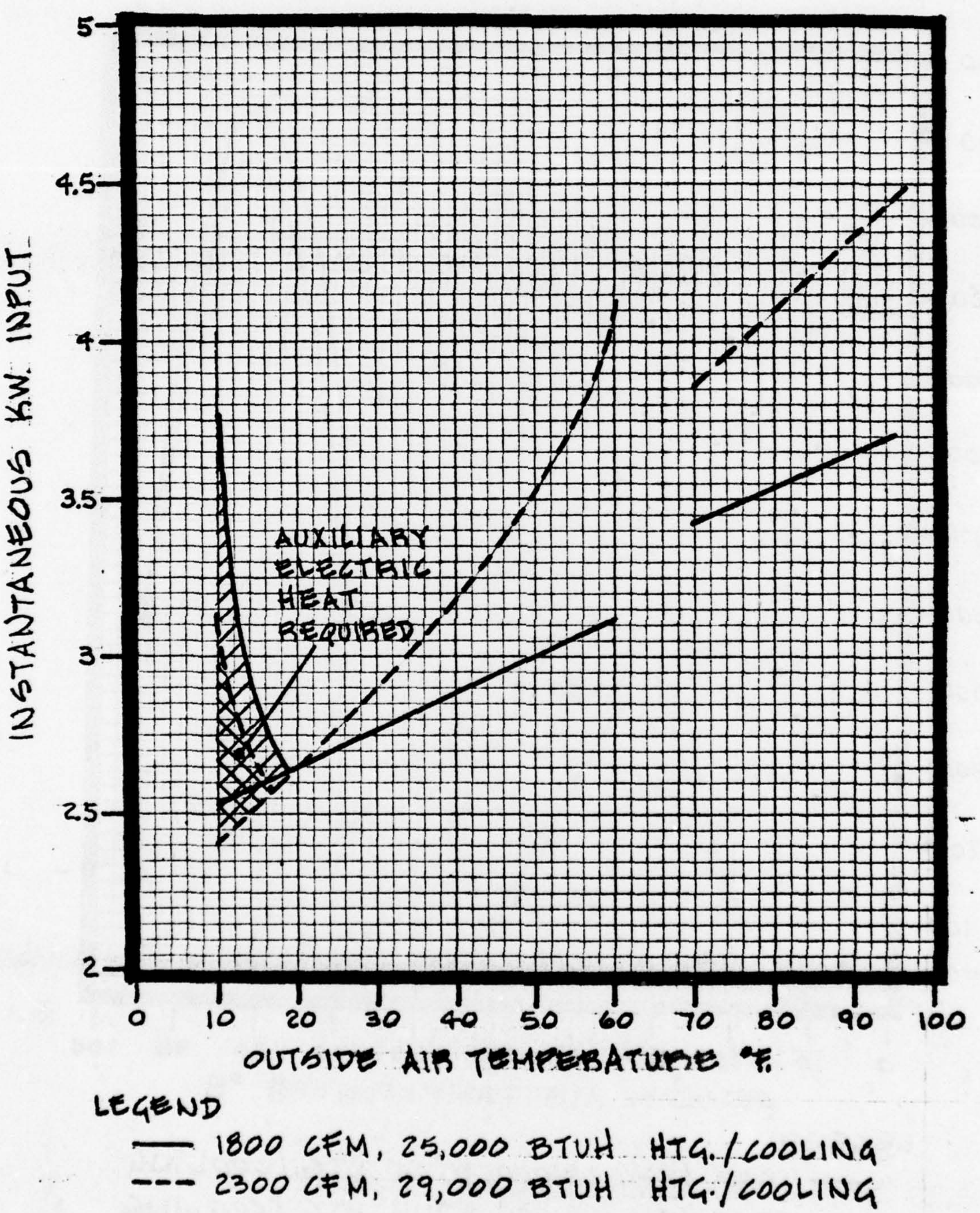


Figure 50. Heat Pump Comparison - **Instantaneous Performance**

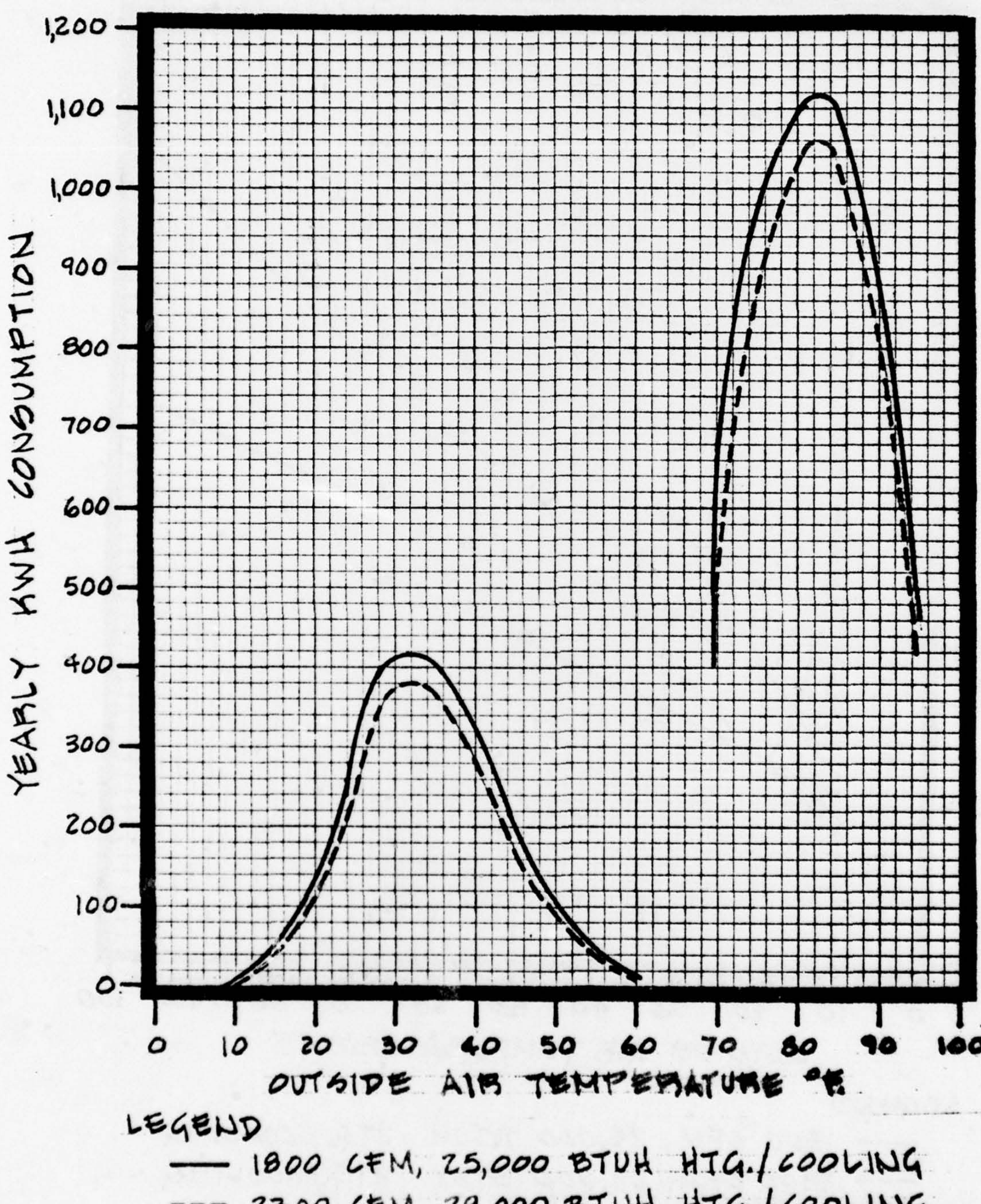


Figure 51. Heat Pump Comparison - Annual Performance

Table 6. Heat Pump Comparison - Energy Consumption

UNIT SIZE	HEATING SEASON			COOLING SEASON	DOMESTIC HOT WATER	TOTAL KILOWATT HOURS	TOTAL COST @ $0.022 per KWH
	Heat Pump Including Fans KWH	Auxiliary Energy KWH	Total KWH	KWH	KWH		
25,000 Btu/Hr 1,800 cfm	2,304	16	2,320	4,951	7,556	17,147	$ 381
29,000 Btu/Hr 2,300 cfm	2,033	3	2,036	4,582	7,556	14,174	$ 315

array as compared to geographical areas where the heating load is dominant. If solar energy were also being used to meet the cooling load in conjunction with an absorption chiller or a Rankine Cycle drive, the optimum size would shift again as a function of the dominant load. The exact combination of optimum collector area and storage volume will be determined during detailed system design in Phase III.

3.8 SYSTEM COMPARISON

In order to develop a comparison between Systems 3 (air) and 6 (liquid), a range of collector areas was superimposed on the basic heat pump system. Typical liquid and air collector efficiency curves were used. System performance was based on average weather conditions including the effect of cloud cover. Collector operating temperature was assumed to average approximately 110°F during the heating season and 135!F during the remainder of the year when the domestic hot water load is dominant. Energy savings were determined for each of the five sizes analyzed. Energy required to operate the solar system was included. The results of this analysis are shown in Table 7.

Further comparison of Systems 3 and 6 was limited to collector areas providing approximately the same energy savings and with annual solar participation in the range of 65-80 percent. Thus, the 100 and 150 square foot liquid systems were compared with the 200 and 250 square foot air systems, respectively. The relationship between the cumulative heating and domestic hot water loads and solar energy use for 67 and 76 percent annual participation is shown in Figure 52. Figure 53 shows the monthly relationship.

Tables 8 and 9 show a cumulative cash flow comparison between the two system sizes mentioned above. This analysis is based on a net increase in energy cost of 5 percent per year after inflation and a present energy cost of $0.0222 per KWH. Table 8 compares the two systems giving approximately 67 percent annual solar participation with the result that the cumulative cash flow turns positive in 22 years for the liquid system and 30 years for the air system. Table 9 compares the two systems giving approximately 76 percent annual solar participation with the result that the cumulative cash flow turns positive in 22 years for the liquid system and 32 years for the air system.

This analysis is based on the cost of existing hardware and could vary significantly if there are any substantial price changes in the collector industry. However, based on today's costs, System 6, the liquid system, is more attractive.

Table 7. System Comparison

	COLLECTOR AREA SQ.FT.	ANNUAL SOLAR PARTICIPATION %	ENERGY REQUIRED FOR HEATING KWH	ENERGY REQUIRED FOR DHW KWH	ENERGY REQUIRED TO OPERATE SOLAR SYSTEM KWH	TOTAL ANNUAL KWH CONSUMED	TOTAL ANNUAL ENERGY REQUIRED BY CONVENTIONAL SYSTEM - HEATING & DHW	ANNUAL SAVINGS KWH	ANNUAL SAVINGS $
SYSTEM NO. 3	100	52	1,058	3,469	1,644	6,171	9,592	3,421	76
	150	62	688	3,041	1,769	5,498	9,592	4,094	91
	200	69	446	2,684	1,784	4,914	9,592	4,678	104
	250	77	278	2,189	1,984	4,451	9,592	5,141	114
	300	82	138	1,902	2,005	4,045	9,592	5,547	123
SYSTEM NO. 6	100	67	582	2,707	1,572	4,861	9,592	4,731	105
	150	76	313	1,992	1,652	3,957	9,592	5,635	125
	200	83	101	1,758	1,646	3,505	9,592	6,087	135
	250	88	0	1,459	1,656	3,115	9,592	6,477	144
	300	91	0	1,061	1,782	2,843	9,592	6,749	150

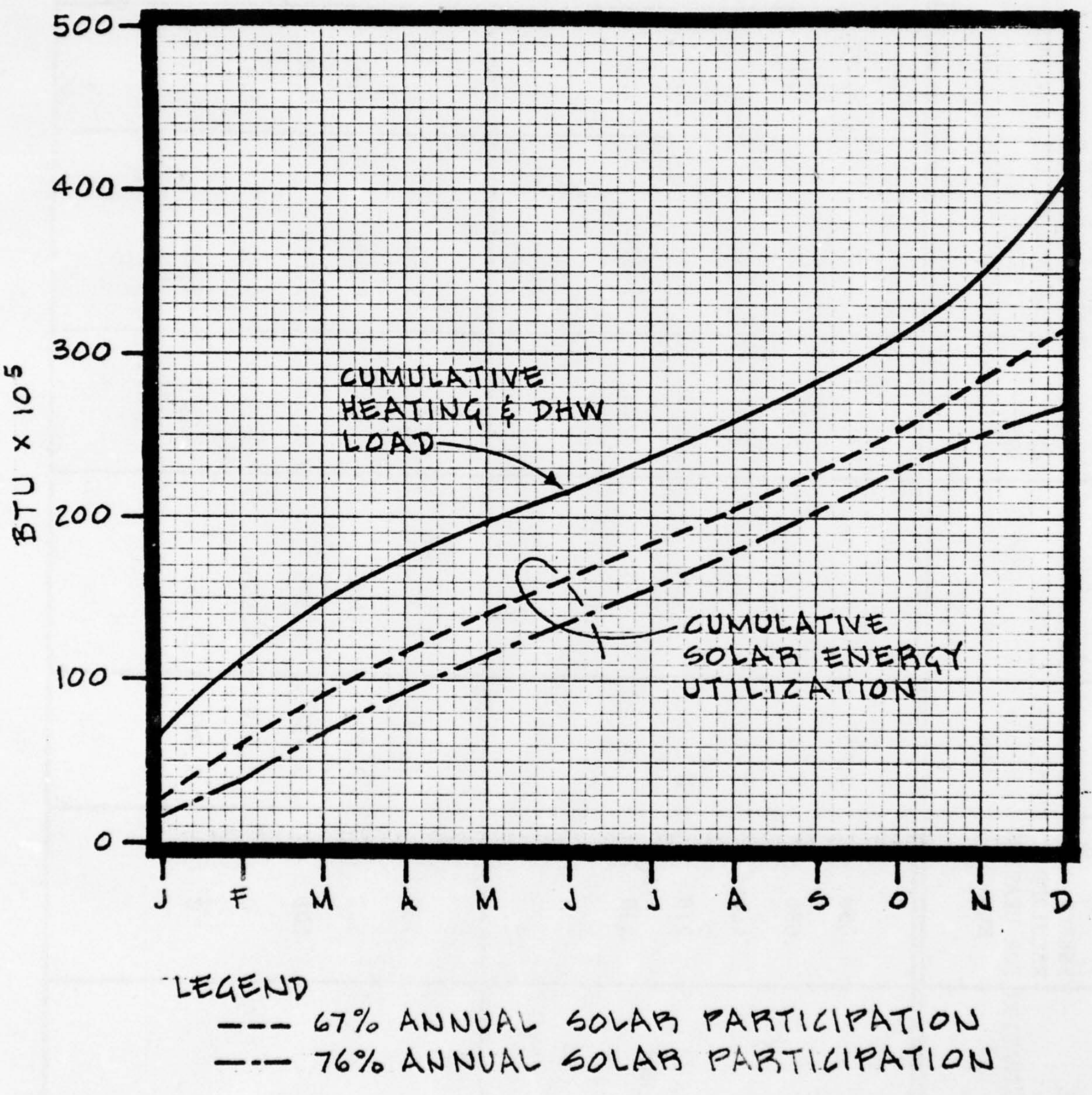


Figure 52. Cumulative Loads and Solar Utilization

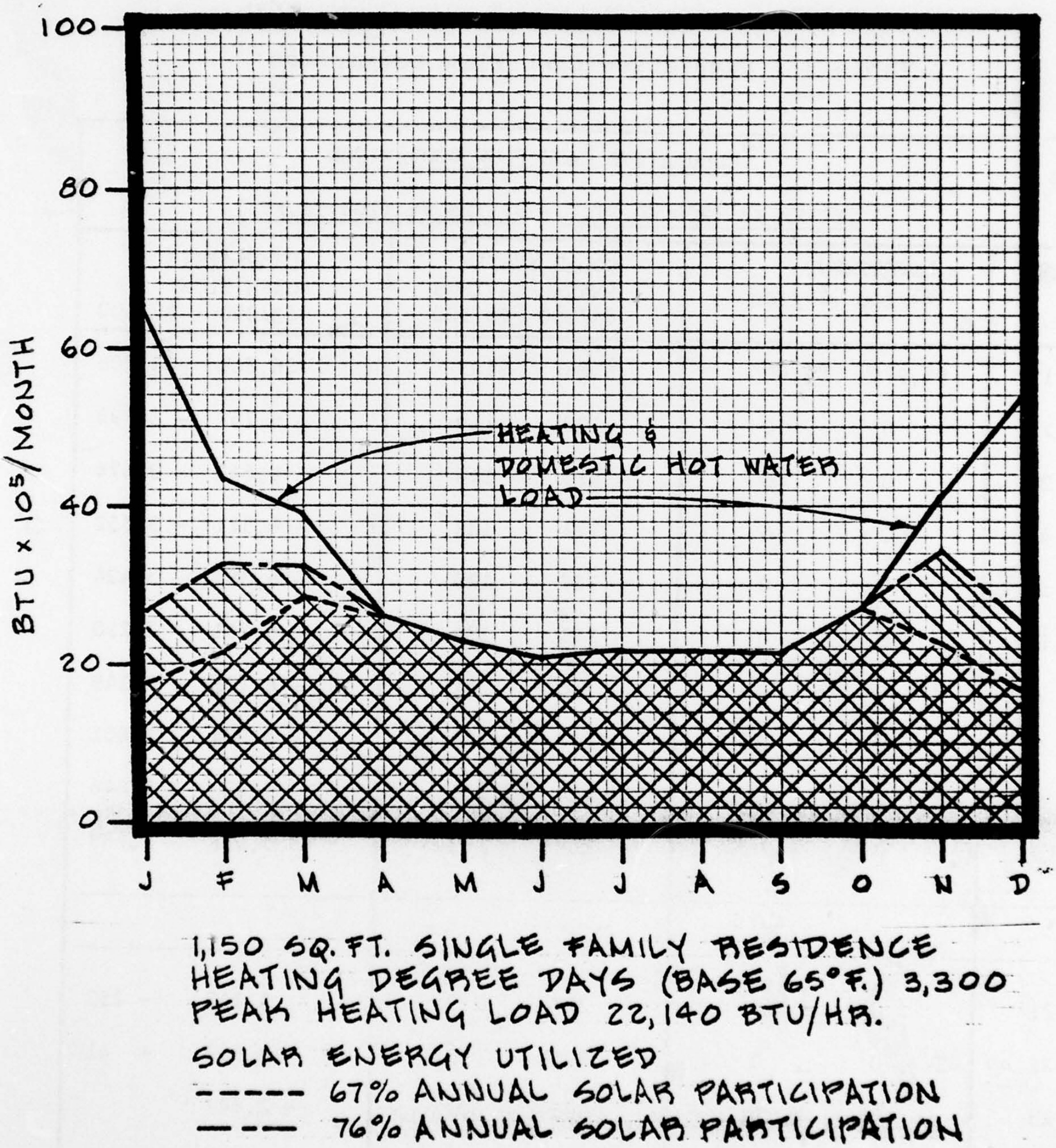


Figure 53. Annual Load Distribution/Solar Participation

Table 8. Cumulative Cash Flow Comparison

CUMULATIVE CASH FLOW COMPARISON

SYSTEM #3 (200 SQ.FT.) - SYSTEM #6 (100 SQ.FT.)

YEAR	INVESTMENT #3/200	#6/100	SOLAR SYSTEM ANNUAL SAVINGS #3/200	#6/100	CUMULATIVE CASH FLOW #3/200	#6/100
1	$6,970	$4,005	$104	$105	-$6,866	-$3,900
2	0	0	109	110	- 6,757	- 3,790
3	0	0	115	116	- 6,642	- 3,674
4	0	0	121	122	- 6,521	- 3,552
5	0	0	127	128	- 6,394	- 3,424
6	0	0	133	134	- 6,261	- 3,290
7	0	0	140	141	- 6,121	- 3,149
8	0	0	147	148	- 5,974	- 3,001
9	0	0	154	155	- 5,820	- 2,846
10	0	0	162	163	- 5,658	- 2,683
21	0	0	277	279	-3,243	- 252
22	0	0	291	293	-2,952	+ 41
23	0	0	305	307	- 2,647	
29	0	0	409		- 469	
30	0	0	429		+ 40	

Table 9. Cumulative Cash Flow Comparison

CUMULATIVE CASH FLOW COMPARISON

SYSTEM #3 (250 SQ.FT.) - SYSTEM #6 (150 SQ.FT.)

YEAR	INVESTMENT #3/250	INVESTMENT #6/150	SOLAR SYSTEM ANNUAL SAVINGS #3/250	SOLAR SYSTEM ANNUAL SAVINGS #6/150	CUMULATIVE CASH FLOW #3/250	CUMULATIVE CASH FLOW #6/150
1	$8,065	$4,755	$114	$125	-$7,951	-$4,630
2	0	0	120	131	- 7,830	- 4,499
3	0	0	126	138	- 7,705	- 4,361
4	0	0	132	145	- 7,573	- 4,216
5	0	0	139	152	- 7434	- 4,064
6	0	0	145	160	- 7289	- 3,904
7	0	0	153	168	- 7136	- 3,736
8	0	0	160	176	- 6976	- 3,560
9	0	0	168	185	- 6808	- 3,375
10	0	0	177	194	- 6631	- 3,181
21	0	0	302	332	- 3993	- 287
22	0	0	318	348	- 3675	+ 61
23	0	0	333	366	- 3342	
30	0	0	469		- 495	
31	0	0	492		- 3	
32	0	0	517		+ 514	

SECTION IV

PHASE IIA

4.1 GENERAL

To allow valid comparison of the solar assisted hybrid heat pump system (System #22) with systems already evaluated in Phase II, the same typical single family ranch-type house with improved thermal characteristics was assumed. This house is illustrated in Figure 30 and is assumed to be oriented so that the solar collectors face true south. The same house will also be used to determine the suitability of System #22 at Scott AFB.

The constraints imposed on system installation by an existing structure were fully addressed in Phase II.

Initially, Systems #3 and #6 were evaluated for collector areas of 100, 150, 200, 250 and 300 square feet. Since the hybrid heat pump system must be capable of operating in the water-to-air mode, the analysis in this report is based on the integration of the hybrid heat pump with a liquid collector system (System #6). With liquid collectors (System #6) it was found that collector areas of 250 square feet and over met nearly the full yearly heating load in Little Rock resulting in minimum heat pump operation and poor economic payback. Final evaluation in Phase II was therefore made for 100 and 150 square feet of collectors only. This same situation will apply to the solar assisted hybrid heat pump and this report will therefore evaluate collector areas of 75, 100, 150 and 200 square feet for Little Rock.

Due to the more adverse winter climate at Scott AFB, larger collector areas must be used to achieve the same solar participation as at Little Rock AFB. Appropriate collector areas will be selected to study the effectiveness of System #22 at Scott AFB.

When making the survey of manufacturers it was found that there are many interpretations of the term "hybrid heat pump."

To obtain comparable answers, a hybrid heat pump is defined as follows:

The heat pump should be a reversible refrigeration system capable of heating and cooling supply air with the same in-duct coil.

In the cooling cycle heat should be rejected through a coil cooled by outdoor air.

In the heating cycle the heat pump should be capable of using either of two independent sources of heat depending on the dictates of the control system.

The two sources of heat should be:

a. Water from a storage tank at temperatures between 55°F and 100°F.

b. Outdoor air at temperatures between 15°F and 60°F.

The heat pump should use an in-duct electric resistance heater for back-up when neither heat source is available.

The unit should provide for automatic defrost of the outdoor air coil.

4.2 OBJECTIVES

The objectives of Phase IIA were:

To canvass industry and identify three potential manufacturers of hybrid heat pump systems.

To select a typical hybrid heat pump system, prepare schematic drawings and tabulate major component costs.

To determine range of available sizes and the operating characteristics in each operating cycle for one selected size.

To make a preliminary design for a solar assisted hybrid heat pump system and tentatively select system components.

To determine the performance effectiveness of the system in heating and cooling a typical house at Little Rock AFB and Scott AFB.

To estimate the construction cost of the system.

To determine the overall economics and comparative cost effectiveness against System 6 at Little Rock AFB.

To determine the suitability of the system to colder climates typified by a Scott AFB location.

4.3 SURVEY OF MANUFACTURERS

A letter of inquiry (Appendix II) was sent to 44 manufacturers. Of the 44 manufacturers canvassed, 12 replied, indicating some interest, but no immediate intention to manufacture a hybrid heat pump. The consensus appeared to be that the market for such a unit is too small to warrant the commitment of R & D funds and manufacturing space which is already allocated to other profitable products. Manufacturers also indicated that they were already fully employed in producing their standard machines.

The original letters of inquiry were sent and answers were received from the corporate offices of the respective companies. While the answers reflected the current corporate policies which are strongly profit motivated, it was felt that more sympathetic replies might be obtained by approaching the R & D divisions directly. Therefore, a telephone survey was conducted of most of the manufacturers, speaking in each case to selected persons in the R & D field.

Positive answers were received from Weil-McClain, Lennox, and York with an expression of interest from Dunham-Bush.

Weil-McClain is currently developing hybrid heat pumps using its standard water-to-air heat pump modified by an additional refrigerant circuit to an outdoor fan and coil unit. All major components are standard "off the shelf" items. At this time, the units are individually assembled in the R & D department and their cost is roughly twice that of a standard production unit. This cost would, however, be reduced with sufficient orders to justify an assembly line approach.

Weil-McClain was willing to work on this project. Subsequently the vice president of Engineering, Dieter Grether, and his assistant, Mike Hughes, met with the contractor. Research data giving performance characteristics over a range of conditions, were provided for each mode of operation. These performance data, illustrated in Figures 54 and 55 for the air-to-air and water-to-air modes, respectively, were used as the basis for this report.

York currently markets a water-to-air heat pump unit under the trade name of "Triton." A number of these units have been converted to hybrid operation although the method is a little different from Weil-McClain. York's approach is to add a liquid/refrigerant heat exchanger and use a standard outdoor air coil with antifreeze as the transfer fluid.

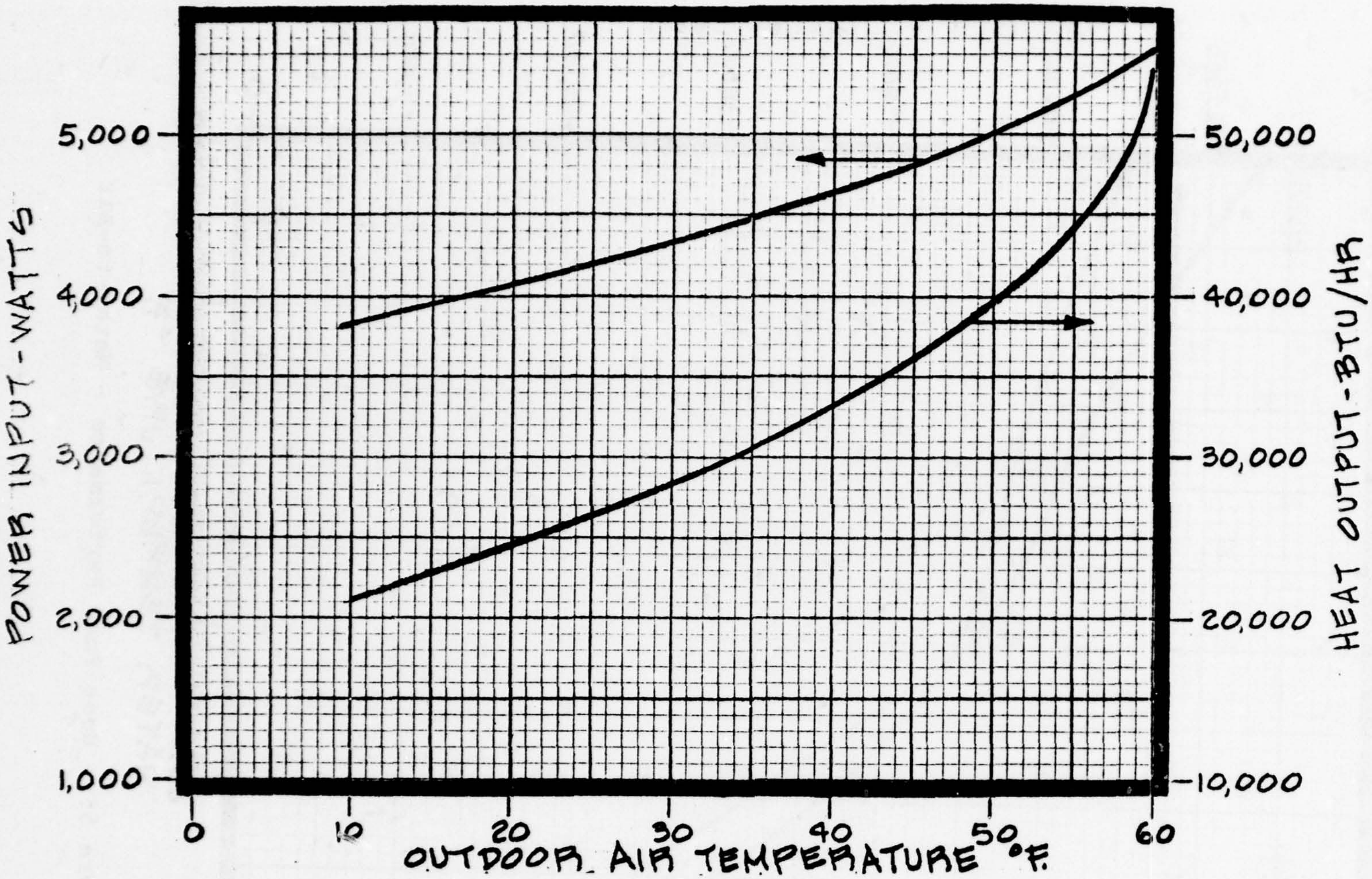


Figure 54. Heat Pump Performance - Air-to-air

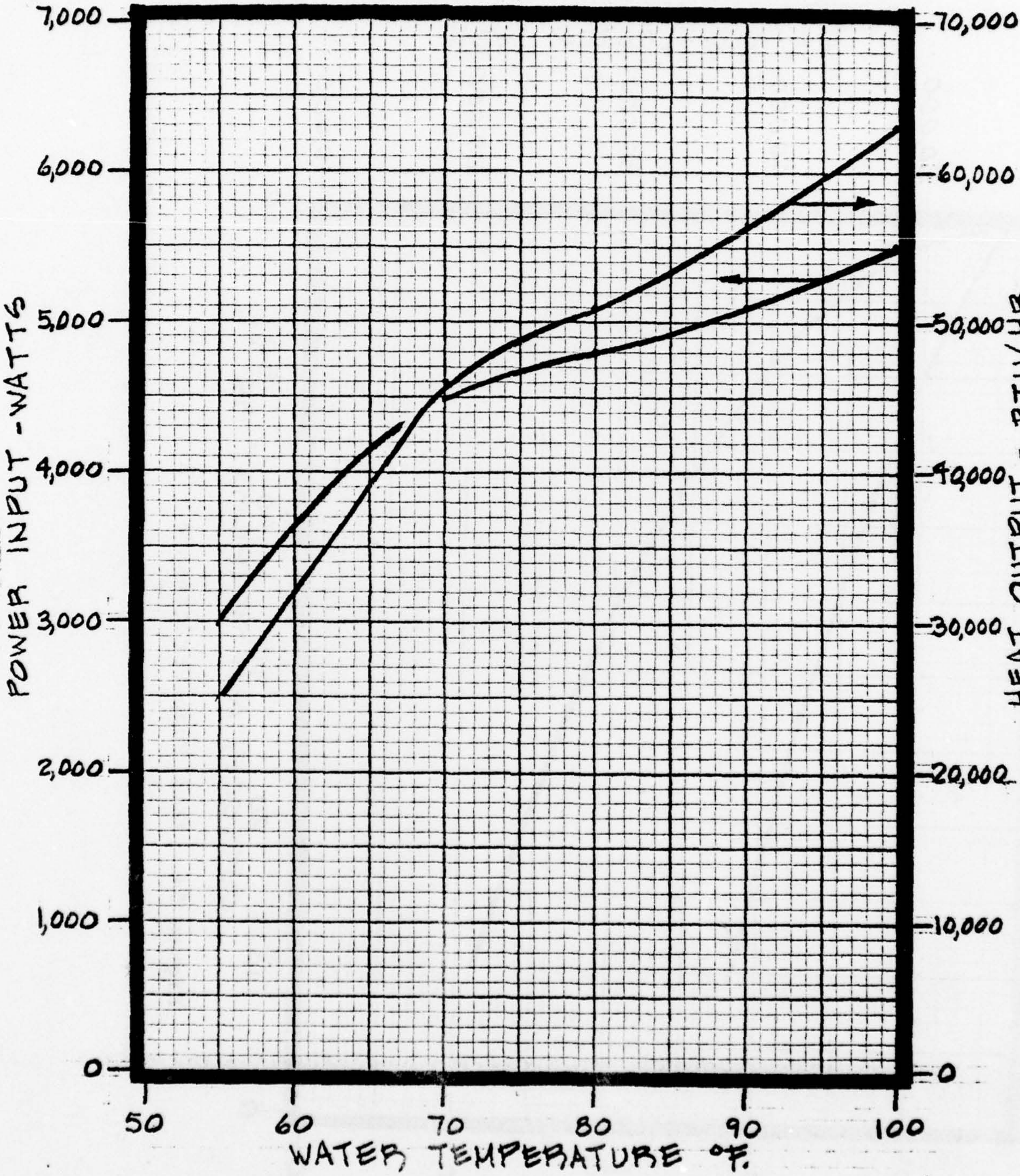


Figure 55. Heat Pump Performance - Water-to-Air

Although this reduces the potential COP on the air-to-air heating cycle, it simplifies the refrigerant system and eliminates difficulties encountered with differing charge requirements of each cycle.

York has some pilot systems installed and will furnish performance data at a later date. The man in charge of hybrid heat pump development is Charles Haley at York's factory in York, Pa.

Lennox defines any heat pumps other than standard production items as hybrid, and some confusion was experienced until the particular definition was outlined. A number of prototype models are assembled and working. Lennox does have one pilot installation in a solar home in Dallas which is instrumented and operated by Texas Power and Light. Data from this system will be available shortly.

In summary, there appears to be a quickening interest in hybrid heat pumps by manufacturers. Weil McClain, Lennox, and York all have operating pilot systems and are willing to market these on a "special" basis at first, and if the market expands, to develop standard products. All manufacturers agree, however, that the selection of the "best" operating mode is not as simple as it first appears and are thinking in terms of developing a microprocessor chip as a basis for the control system.

4.4 METHODOLOGY

The methodology in compiling this report was first to canvass heat pump manufacturers and identify at least three capable of making a hybrid heat pump system using standard components. The results of this effort have been described in Section 4.3.

To establish a base for comparison, the same typical building was assumed with conventional split heat pump system that was used in the Phase II report. Phase II report calculations were re-used to establish the yearly load profile and yearly energy consumption for this building located in Little Rock, Arkansas.

Average weather data for Scott AFB was determined, using AFM 88-8 as the source for hourly temperatures, the ASHRAE tables as the source for insolation, and the Climatic Atlas for data regarding cloud cover.

A yearly load profile and base yearly energy consumption was then calculated for the same house and conventional system at Scott AFB. The load profile was used to select the heat pump capacity with the ruling factor being the peak heating load in the air-to-air configuration.

A preliminary design for a solar assisted hybrid heat pump system was established and control logic formulated to achieve the desired modes of operation. Detailed individual flow diagrams were prepared for each mode of operation together with a control diagram using standard control hardware. It became apparent that a conventional control system using fixed parameters such as storage tank temperature, outdoor air temperature, etc. would not make full use of the system potential but must be designed for the best compromise. This conventional type of control system has been presented in this report.

The ideal control system is one that could make mode change decisions based on load trend, weather trend, and time of day and year. At present, this indicates the use of a mini-computer which is cost prohibitive for a single family dwelling. Rapid advances, however, are being made in the design and manufacture of micro-processor chips and large scale integration. This technology could result in a low cost mini-computer suitable for residential control systems. Although this is outside the scope of this report, it should be subjected to further study.

Based on the fixed parameters of the control system, the load profile and the weather profile which determined the mode of operation, the energy use was calculated on an hourly basis for one typical year for Little Rock AFB. These calculations were repeated for different collector areas.

Many other variables could, of course, have been introduced but would have increased the calculation runs exponentially. Storage capacity, for instance, was held constant at 2.5 gallons/square foot of collector area. Experience indicates that this gives the best practical compromise between the conflicting desirable qualities of large thermal storage for cloudy day carry-over and rapid temperature rise to allow direct use of solar heat. The sensitivity of system performance was tested to a range of storage volumes of 1.5 to 4 gallons/square foot for a typical sunny January day at Little Rock AFB. The results of this test are discussed in Section 4.8.

The cost of the hybrid heat pump was furnished by Weil McClain. Cost of the overall solar assisted hybrid heat

pump system was estimated using manufacturers' prices and costs from "Means Building Construction Costs" and the "Bradford Price Book."

Based on the calculated energy consumption and cost of installation, the economic worth of the system for various collector areas at Little Rock AFB was assessed. These costs are directly comparable to those established in Phase II for Systems 3 and 6.

The performance of the hybrid system at Scott AFB was based on a detailed hourly analysis for the month of January for various collector areas.

4.5 SYSTEM CHARACTERISTICS

The hybrid heat pump unit, shown schematically in Figures 56 through 58, is capable of operating in three heating cycles and one cooling cycle.

Air-to-Air Heating (Figure 56)

In this cycle, hot refrigerant gas from the compressor is directed by the reversing valve to the indoor coil where it condenses and gives up heat to the supply air. The hot condensed liquid refrigerant from the indoor coil is directed through the distribution valve to an expansion valve which reduces the liquid pressure and by partial vaporization reduces the liquid temperature. This cool liquid refrigerant is then directed by the control valves through the outdoor air coil where it absorbs heat from the warmer outdoor air and changes state to a cool vapor. This cool vapor is then drawn into the compressor crankcase to complete the cycle. Additional components such as receivers, filter/driers, etc., may be necessary and would be added as required by the manufacturer.

Water-to-Air Heating (Figure 57)

This cycle is the same as the air-to-air cycle previously described except that liquid refrigerant is directed through the water to refrigerant heat exchanger instead of the outdoor air coil. The water-to-air heat exchanger is supplied with warm water which acts as the heat source to vaporize the liquid refrigerant.

Electric Resistance Heating

In this cycle the compressor is turned off and the supply air is heated by a direct electric resistance coil in the air stream. This mode of heating is for back-up only

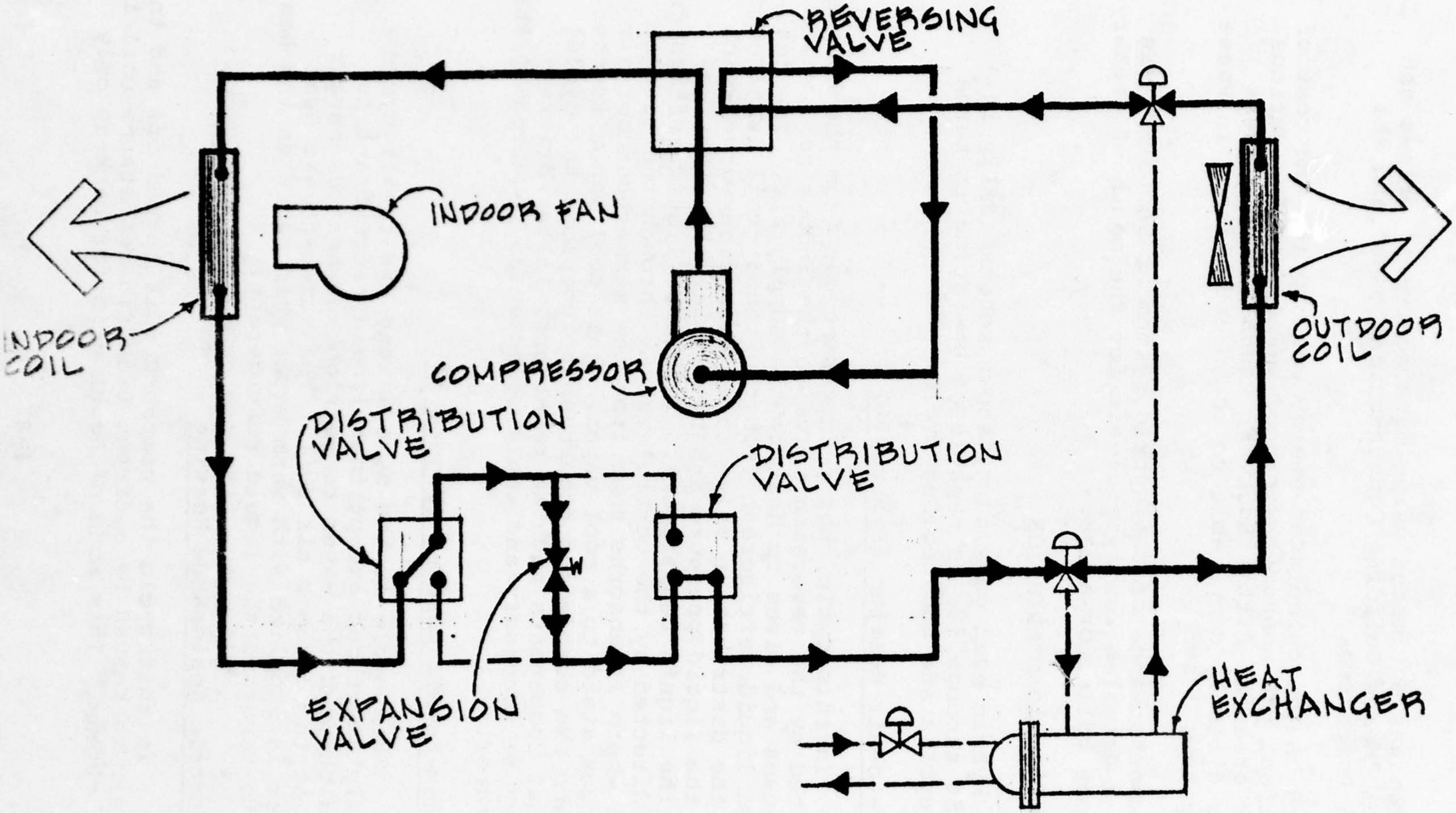


Figure 56. Heat Pump Schematic - Air-to-Air

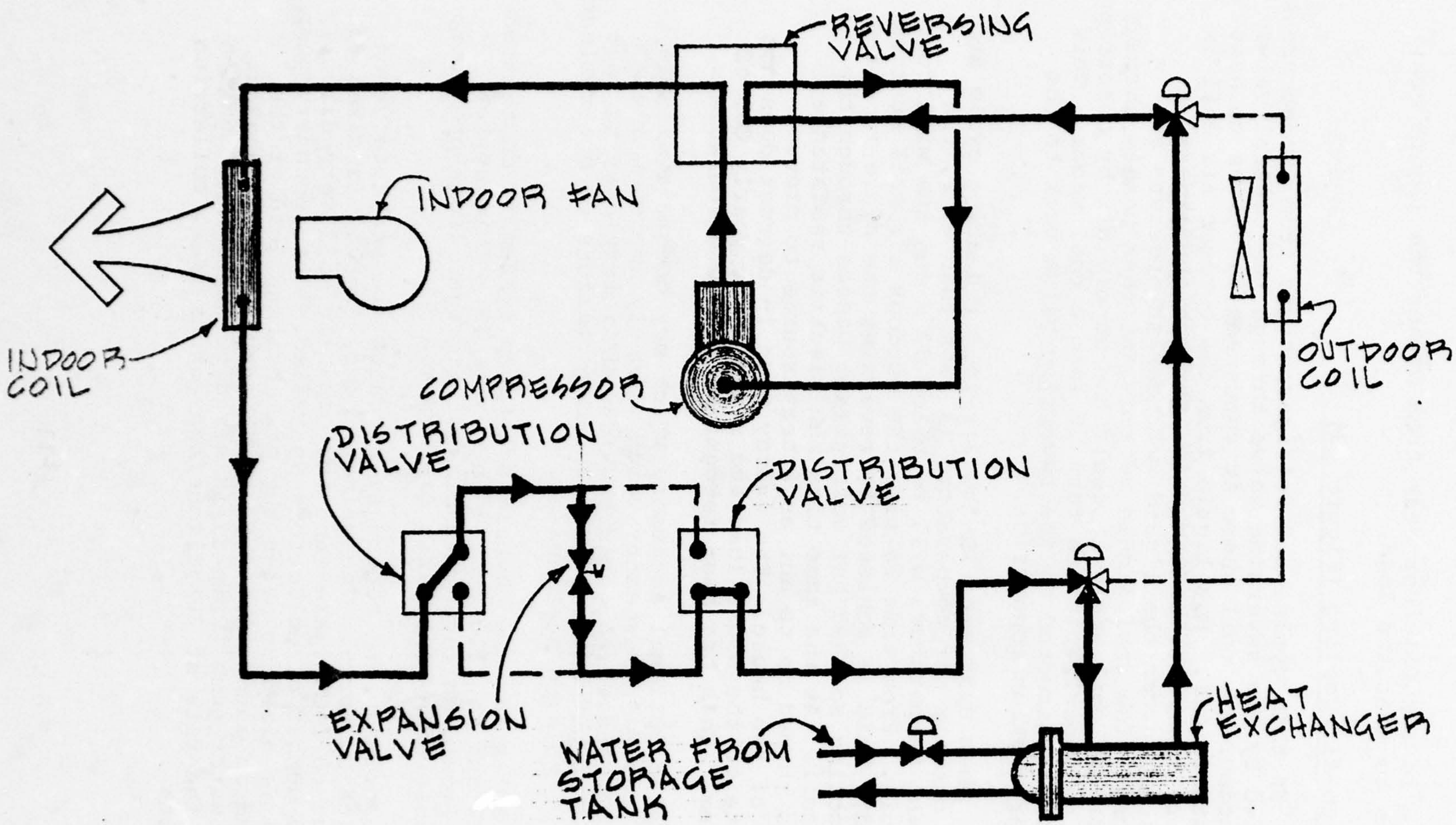


Figure 57. Heat Pump Schematic - Water-to-Air

when there is insufficient heat from either the liquid or air source to meet the load.

Air-to-Air Cooling (Figure 58)

In this cycle, hot refrigerant gas from the compressor is directed by the reversing valve through the control valve to the outdoor air coil where it condenses and gives up heat to the outdoor air. Hot liquid from the outdoor air coil is directed by the distribution valve through the expansion valve which reduces the liquid refrigerant pressure and temperature. The cool liquid refrigerant then passes through the indoor air coil where it cools the supply air by absorbing the heat of vaporization to turn it into a cool vapor. This cool vapor is directed by the reversing valve back to the compressor suction connection.

When operating in the air-to-air heating cycle and when the outdoor air temperature is less than 40°F, the refrigerant temperature will be below 32°F and ice will form on the coil. From time to time the outdoor air coil must be defrosted. This is achieved by reversing the cycle to air-to-air cooling so that hot refrigerant passes through the outdoor coil. At the same time the electric resistance heater is turned on to act as a heat source to prevent cooldown of the house. The energy used in defrost does not contribute to the house load and reduces the yearly COP of the system. This has been recognized in the calculations.

An optional accessory which may become available is a hot gas de-superheater which slightly improves the cooling COP and which could provide sufficient heat to meet the domestic hot water load. The calculations do not reflect the use of this de-superheater.

The solar assisted hybrid heat pump system is designated System 22 and is illustrated in Figure 59. This system employs liquid collectors and water storage, frost protection being achieved by automatic drain down.

The primary and secondary circuits are separate, each having its own pump. The primary pump (Pump 1) is sized to match the collector area and its capacity will vary directly with the number of collectors installed. The secondary pump (Pump 2) is sized to match the peak heating load of the house and is independent of variations in collector area. The hot water pump (Pump 3) is sized to heat the 80 gallon tank at the rate of 20 gallons/hour under ideal collection conditions.

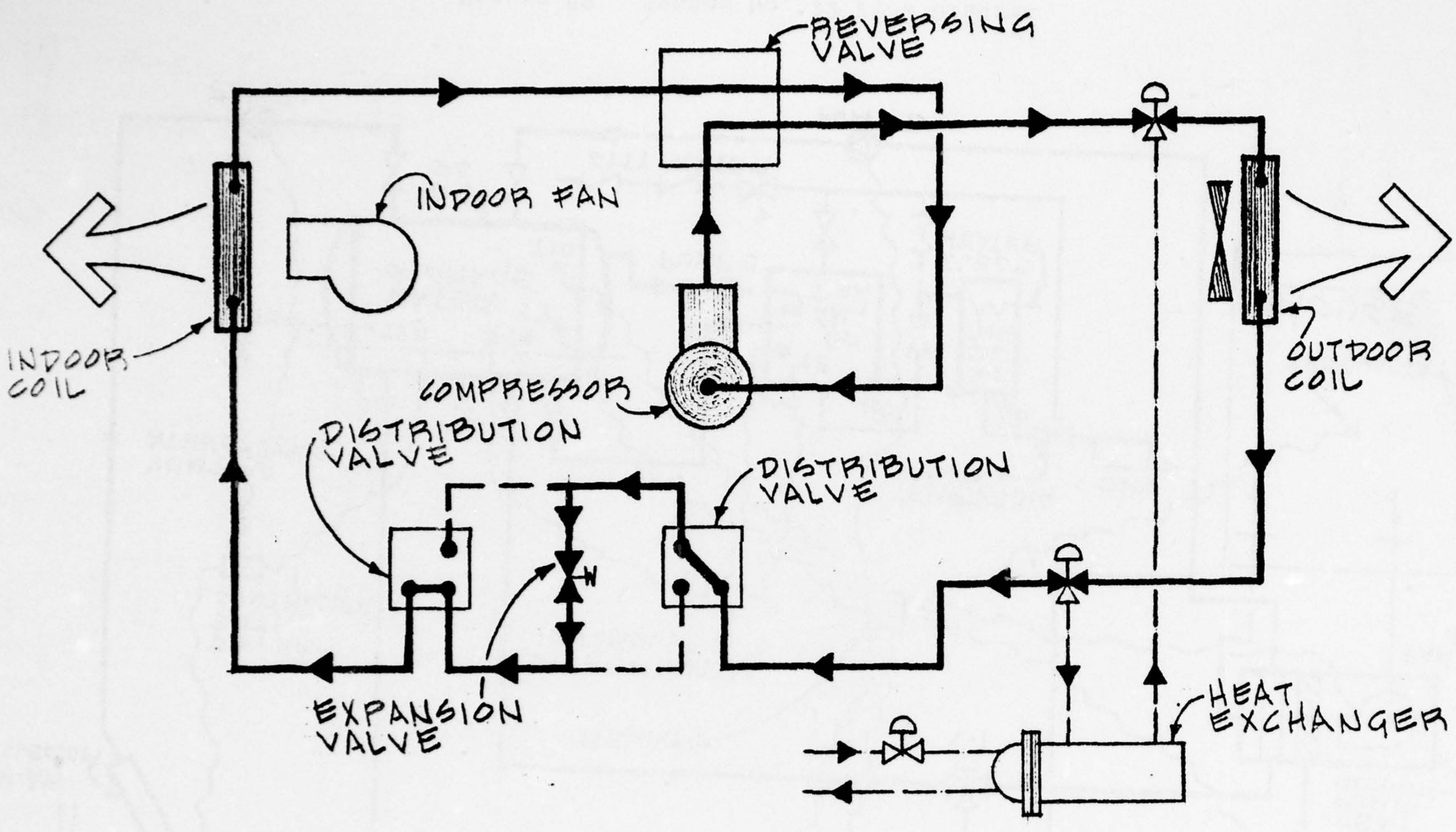


Figure 58. Heat Pump Schematic - Cooling

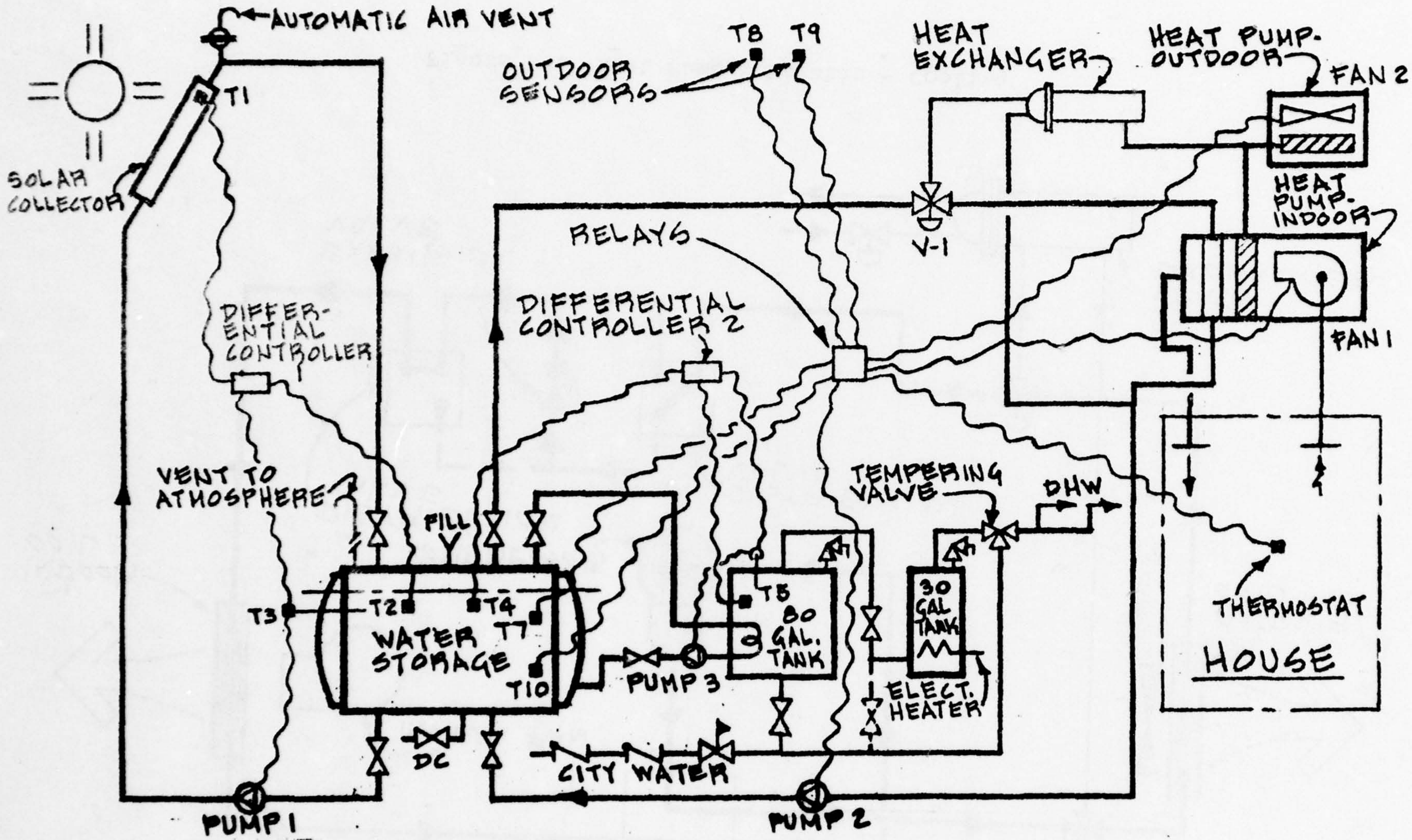


Figure 59. System No. 22 Flow Diagram

The secondary heating circuit supplies hot water to a coil in the air discharge from the heat pump indoor unit. This coil will heat the supply air to the house whenever solar storage water is greater than 100°F.

System #22 control logic is shown in Figure 60 and the control diagram is shown in Figure 61. The controls will operate the system in the following modes:

<u>Solar to Storage (Figure 40)</u>

This mode is for summer and winter operation whenever solar energy is available and the thermal storage tank temperature is less than 180°F. The collector plate temperature and the storage temperature are measured by T_1 and T_2, respectively. When T_1 exceeds T_2 by 20°F, pump 1 will start. The collectors will be filled with water and flow established. Pump 1 will continue to run as long as T_1 exceeds T_2 by at least 3°F. Limit stat T_3 will shut down the solar energy system if the storage temperature exceeds 180°F.

<u>Direct Solar Heating (Mode A - Figure 62)</u>

The secondary solar heating circuit will operate independently of the primary collection circuit and will be controlled by the space thermostat. When the space thermostat falls to 60°F and the storage tank temperature is above 100°F, pump 2 and fan 1 will start and valve V1 will be positioned to direct water through the coil in the supply duct. This mode will continue to operate on demand until the tank storage temperature falls below 100°F.

<u>Heating by Water/Air Heat Pump (Mode B - Figure 63)</u>

When the space thermostat falls below 69°F, the storage tank temperature is between 100°F and 55°F and the outdoor air temperature is less than 40°F, Mode B will be selected. Pump 2 and fan 1 will start and valve V1 will be positioned to direct water through the water/refrigerant heat exchanger and the heat pump compressor will run to upgrade the temperature to heat the supply air in the indoor coil. This mode will continue to run on demand until one of the following conditions occur:

a. Outdoor temperature rises above 40°F.

b. Storage temperature rises above 100°F.

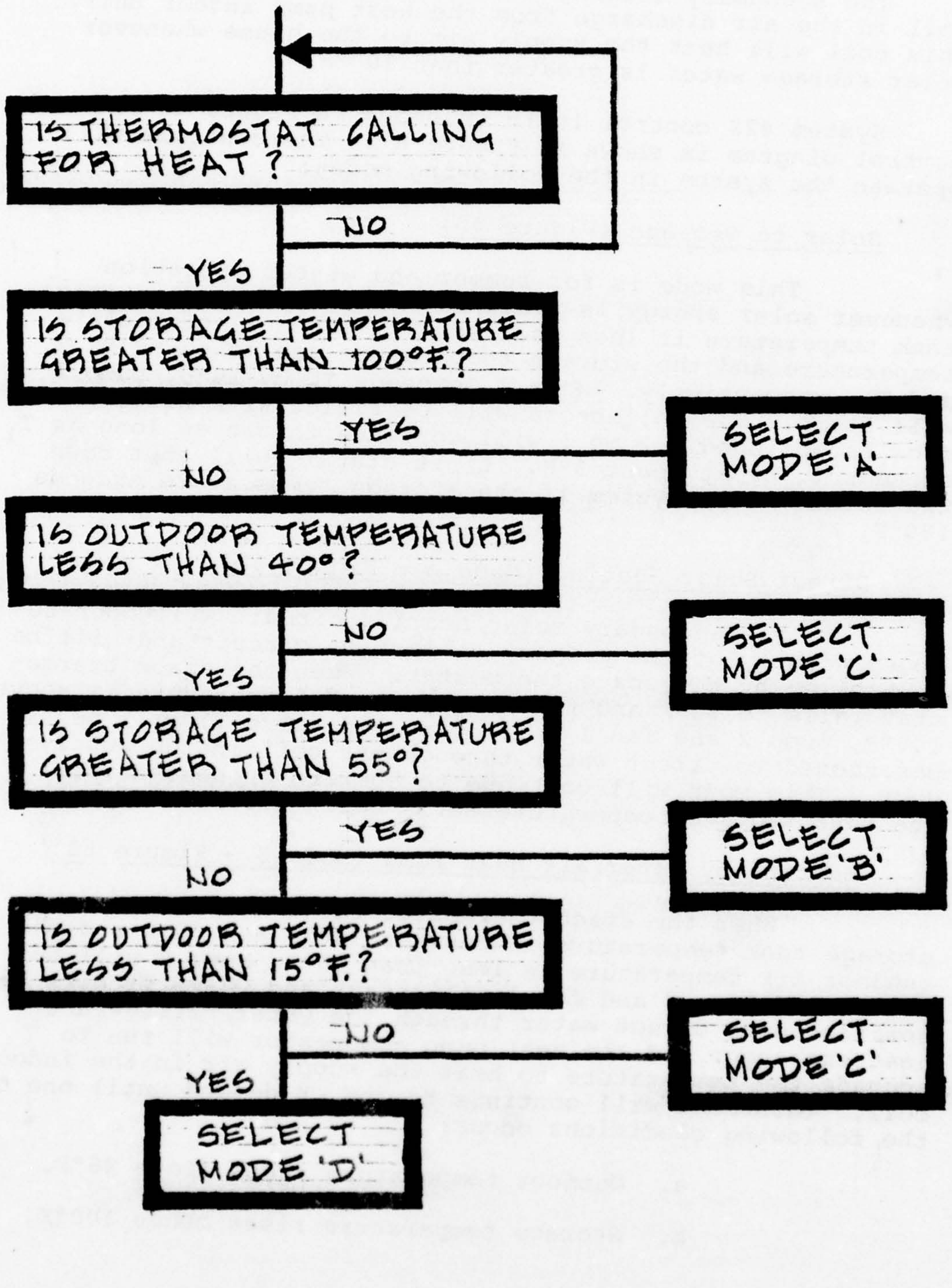


Figure 60. Control System Logic

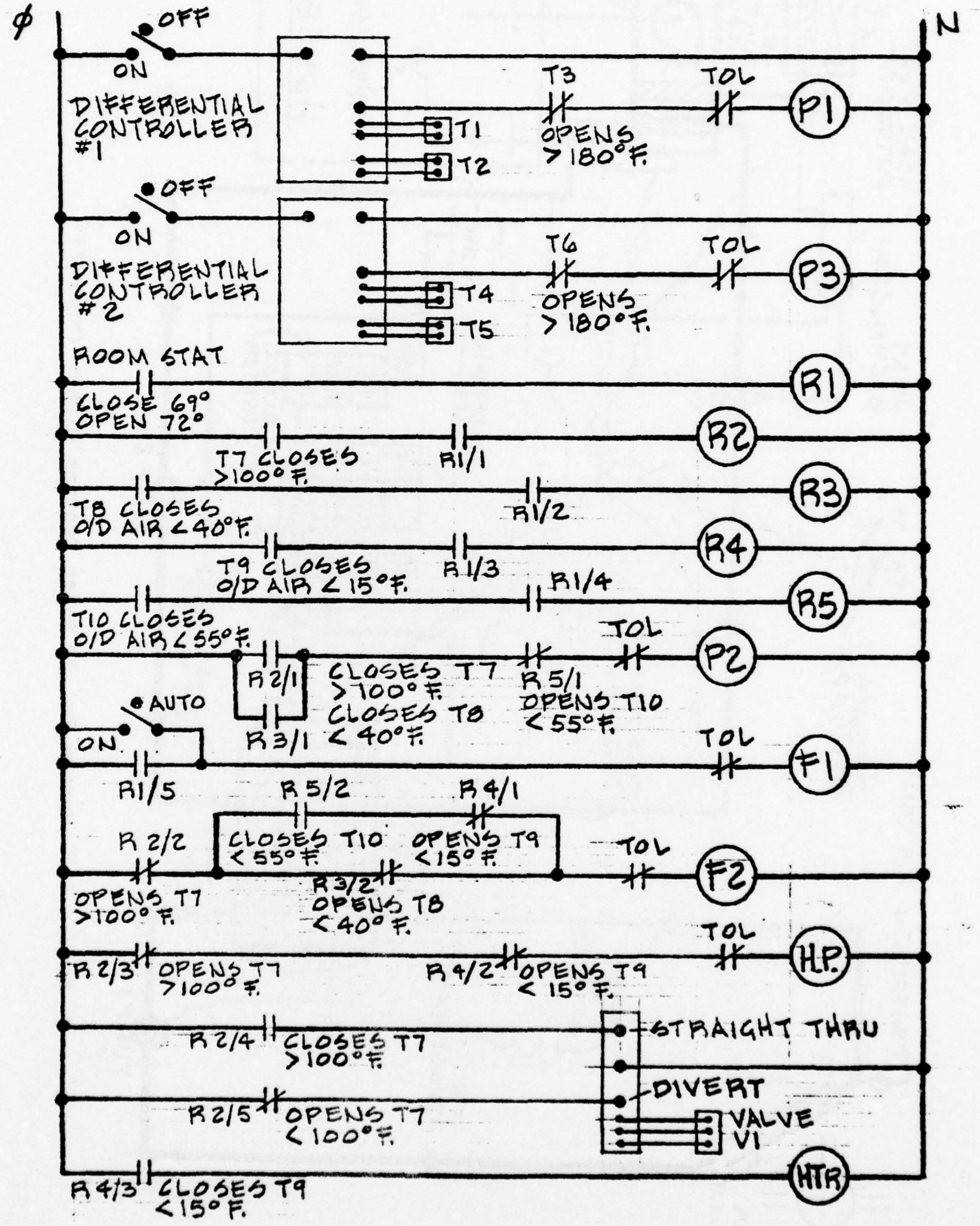


Figure 61. Control System Schematic

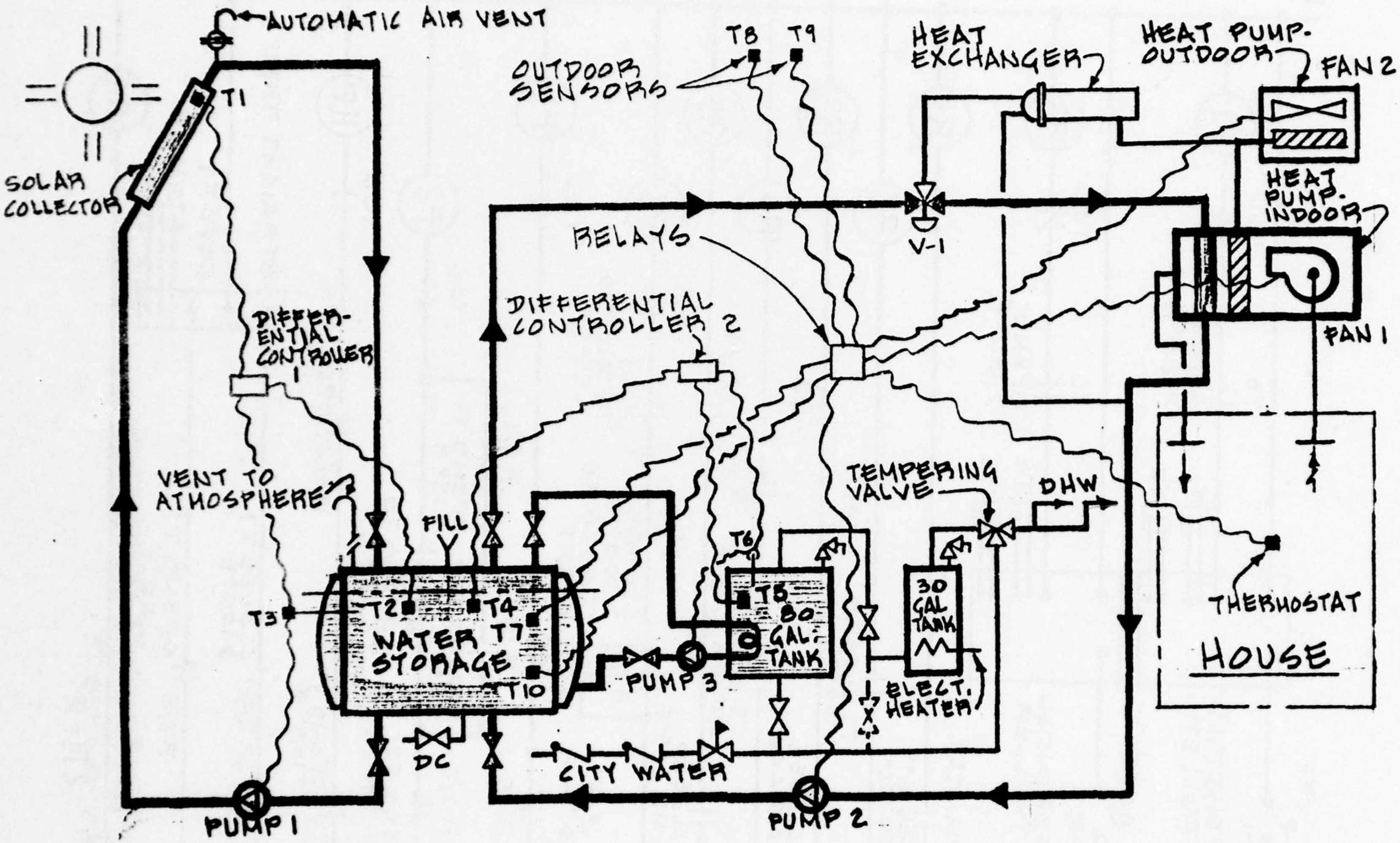


Figure 62. System No. 22 Solar Heating Direct (Mode A)

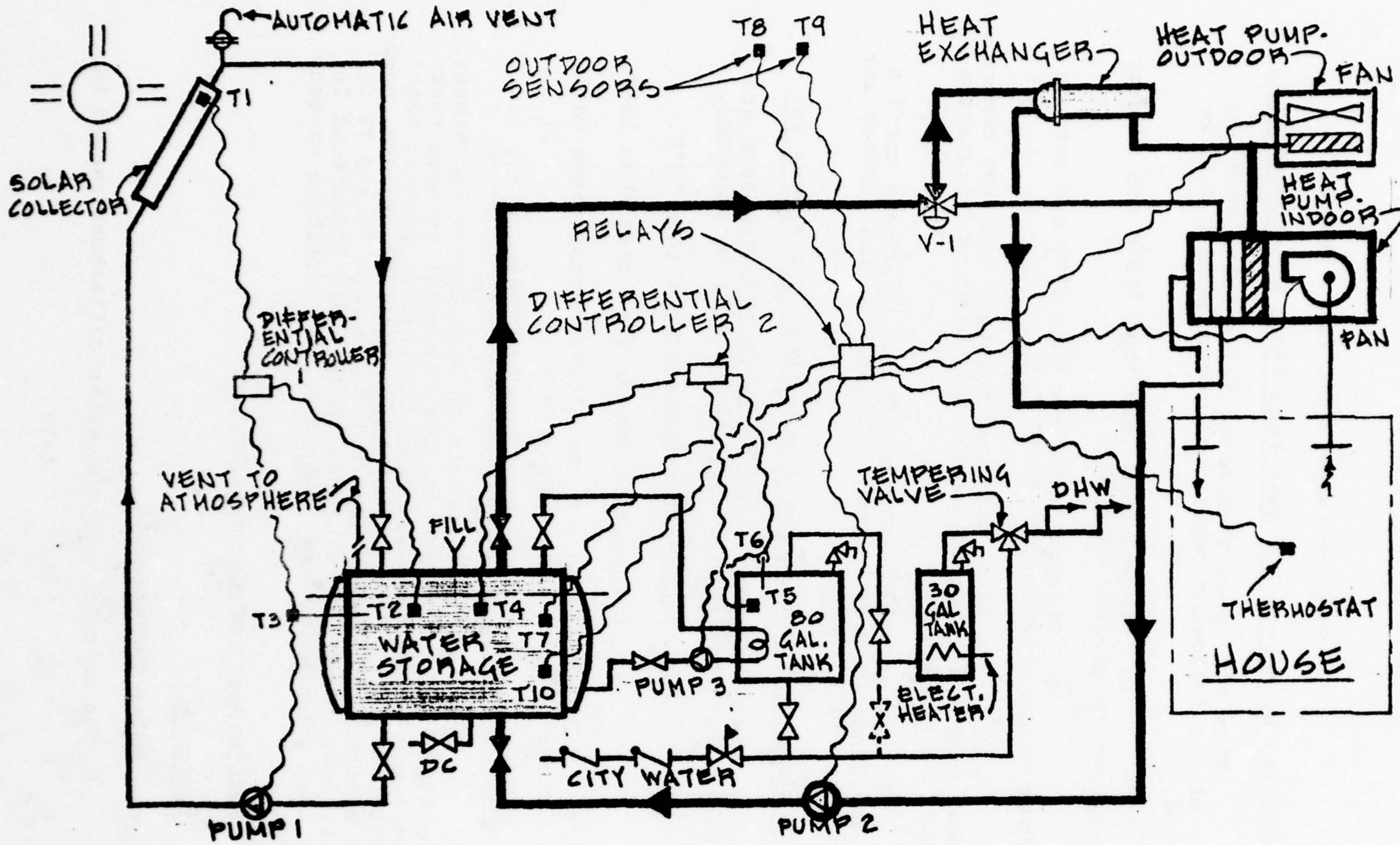


Figure 63. System No. 22 Solar Assisted Water-to-Air Heat Pump (Mode B)

c. Storage temperature falls below 55°F.

Heating by Air-to-Air Heat Pump (Mode C - Figure 64)

When the space thermostat falls below 69°F and either of the following conditions occur, Mode C will be selected.

a. Storage tank less than 100°F and outdoor air temperature greater than 40°F.

b. Storage tank less than 55°F and outdoor air temperature greater than 15°F.

Fan 1 and fan 2 will run together with the compressor. The heat pump will operate in the air-to-air configuration to heat the supply air.

This mode will continue to run on demand until a more advantageous mode becomes available or the outdoor air drops below 15°F.

Heating by Electric Resistance (Mode D - Figure 65)

When the space temperature falls below 69°F and the following conditions occur, Mode D will be selected.

a. Storage temperature less than 55°F.

b. Outdoor air temperature less than 15°F.

Fan 1 will run and the electric resistance coil will energize to heat the supply air.

Solar Domestic Hot Water Generation (Figure 66)

This mode will operate in both Summer and Winter. Differential controller 2 compares the solar storage tank temperature (T4) with the domestic hot water preheat tank temperature (T5). When T4 exceeds T5 by 10°F or more, pump 3 is started. When the differential between T4 and T5 is 3°F or less, pump 3 is stopped. High limit thermostat T6 will stop pump 3 when the preheat tank temperature exceeds 180°F.

4.6 PRELIMINARY EQUIPMENT SELECTIONS

System 6

a. Solar Collectors

The selection of the solar collectors should be

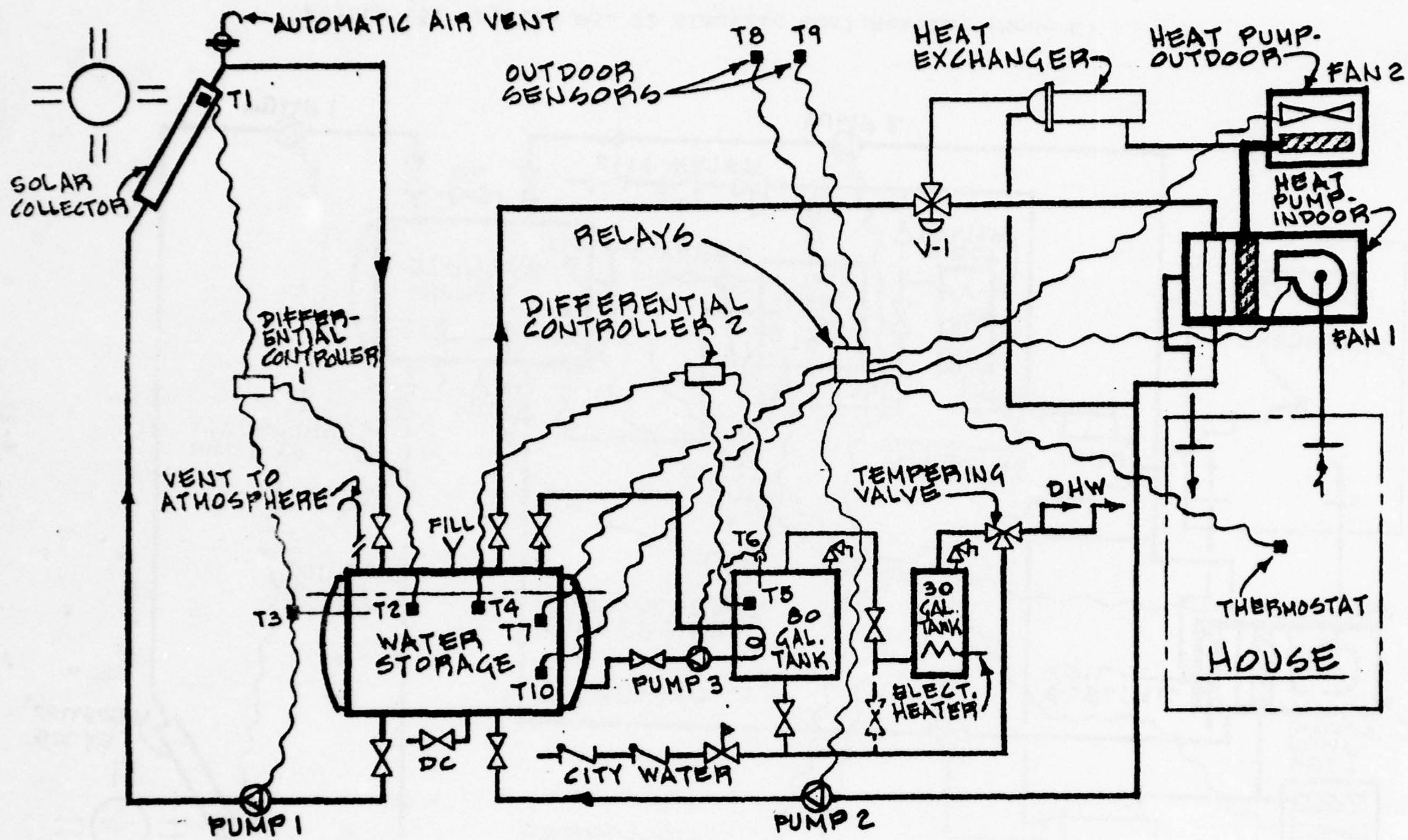


Figure 64. System No. 22 Air-to-Air Heat Pump (Mode C)

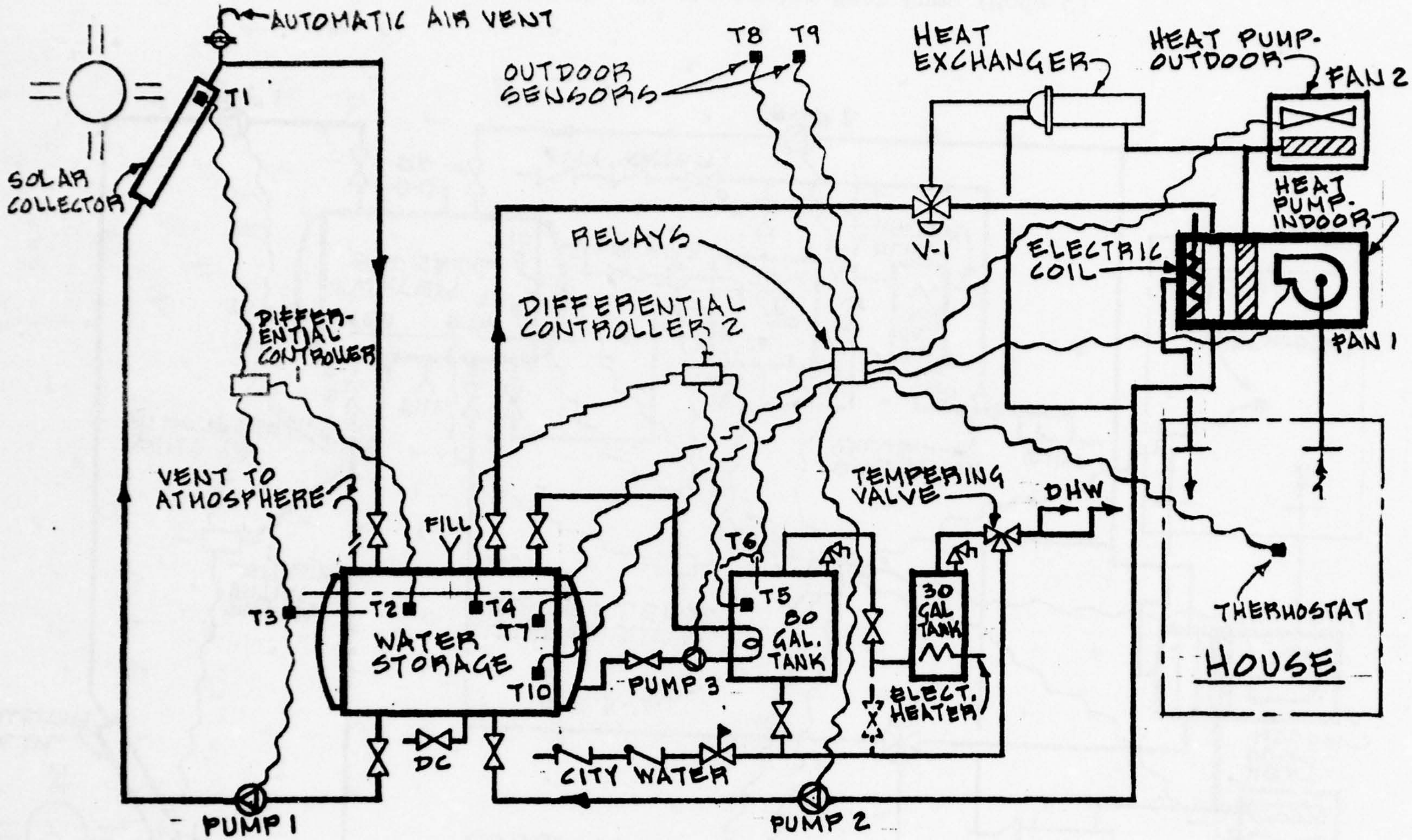


Figure 65. System No. 22 Electric Coil Heating (Mode D)

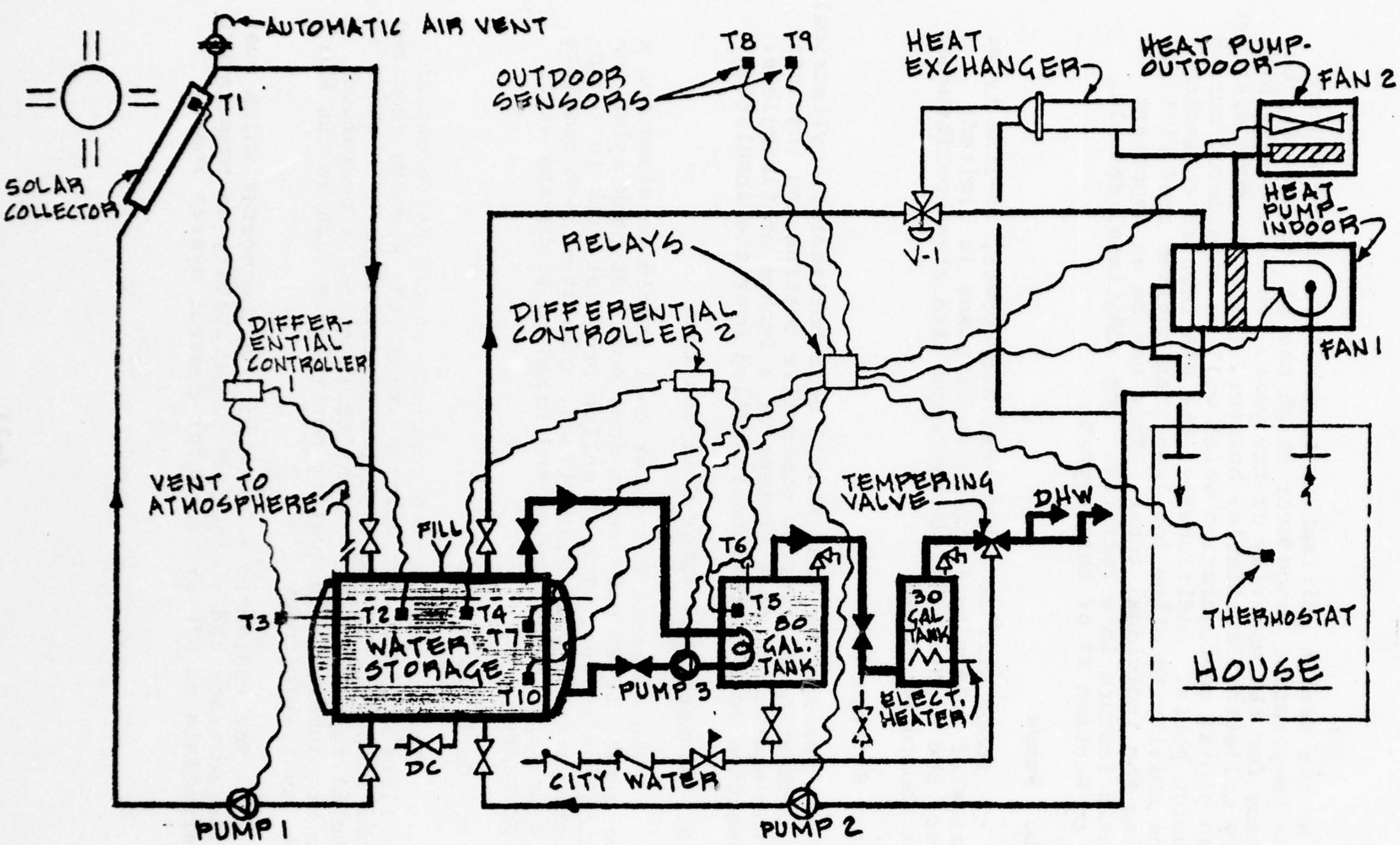


Figure 66. System No. 22 Solar Domestic Hot Water

based on the design utilization temperature of the system and the temperature range over which the collectors will be operating for the majority of the year. There are many high quality collectors available; however, there are some distinctions such as piping configuration which will be considered during the design phase. Additionally, the KTA tubular collector has the ability to allow for less than optimum roof tilt by rotating the individual tubes. This can be an important aesthetic feature in a retrofit. For additional details, refer to Section II of this report.

b. Pumps

Pumps will be fractional horsepower, in-line type. The range of selection in these small sizes is limited; however, the highest efficiency pumps with steep performance curves should be chosen.

c. Storage Tank

The storage tank should be a horizontal, cylindrical steel tank with an interior protective coating and insulated with the equivalent of a minimum of 3 inches of fiberglass. A protective cover should be installed over the insulation.

d. Domestic Hot Water Coil and Tank

The domestic hot water coil should be sized for a recovery rate of 80 gallons in four hours at peak solar insolation conditions. The 80 gallon pre-heat tank is a conventional steel shell, glass lined, insulated tank complete with a pressure relief valve and rated to withstand water main pressure.

e. Controls

The control system consists of two differential controllers designed for solar systems, the sensors required for their operation, and a series of five relays as shown in Figure 61 to accomplish the shifts in modes of operation. There are several high quality controllers such as the Rho Sigma, available today.

The residence will require a thermostat which must provide accuracy and consistent repeatability as temperature differentials of 3°F are used for control system input.

f. Heat Pump

The heat pump can be either a unitary or split hybrid type as described in Section 3.5 and illustrated in Figures 56 through 59. We have shown a split system in the schematic diagrams.

g. Piping, Valves, Fittings, etc.

Piping should be copper, insulated with the equivalent of at least 1 inch of fiberglass. Valves, air vents, and other fittings should be of good quality to give reliable performance.

4.7 SYSTEM CONSTRUCTION COSTS

The system construction costs are based on the tentative equipment selections described in Section 3.5 and on the assumed typical house plan.

Equipment, installation and construction costs were obtained from manufacturers, Means Construction Costs 1977 and other current cost data sources.

The hybrid system was priced for four collector areas ranging from 75 to 200 square feet.

The breakdown showing total estimated installed costs for System #22 is tabulated in Table 10.

4.8 BUILDING LOAD AND SYSTEM PERFORMANCE

The building loads developed for an 1150 square foot single family residence in Little Rock have been used for this report. Loads for Scott AFB were developed in a similar manner based on the hourly temperature distribution from AFM 88-8. Heating loads have been modified by the effect of internal and solar gains. The annual heating, cooling, and domestic hot water load distribution for Scott AFB is shown in Figure 67. Figure 49 from Section III (Phase II) shows the load profile for Little Rock. Peak heating and cooling loads are the same, however, the hourly temperature distributions are substantially different, resulting in a greater annual heating load at Scott AFB and a greater annual cooling load at Little Rock AFB.

Heat pump selection was based on the performance characteristics provided by Weil McClain. The heat pump was sized

Table 10. System No. 22 Estimated Costs

ITEM	COLLECTOR AREA			
	75 FT^2	100 FT^2	150 FT^2	200 FT^2
Collectors	$ 825	$ 1,100	$ 1,650	$ 2,200
Storage Tank	240	330	470	625
Primary Pump	125	140	150	170
Primary Pipe	400	450	500	600
Hot Water Pump	120	120	120	120
Hot Water Tanks	425	425	425	425
Secondary Pump	175	175	175	175
Secondary Pipe	550	550	550	550
Heat Pump (Incl. Coil)	2,185	2,185	2,185	2,185
Controls	450	450	450	450
TOTAL	$ 5,495	$ 5,925	$ 6,675	$ 7,500

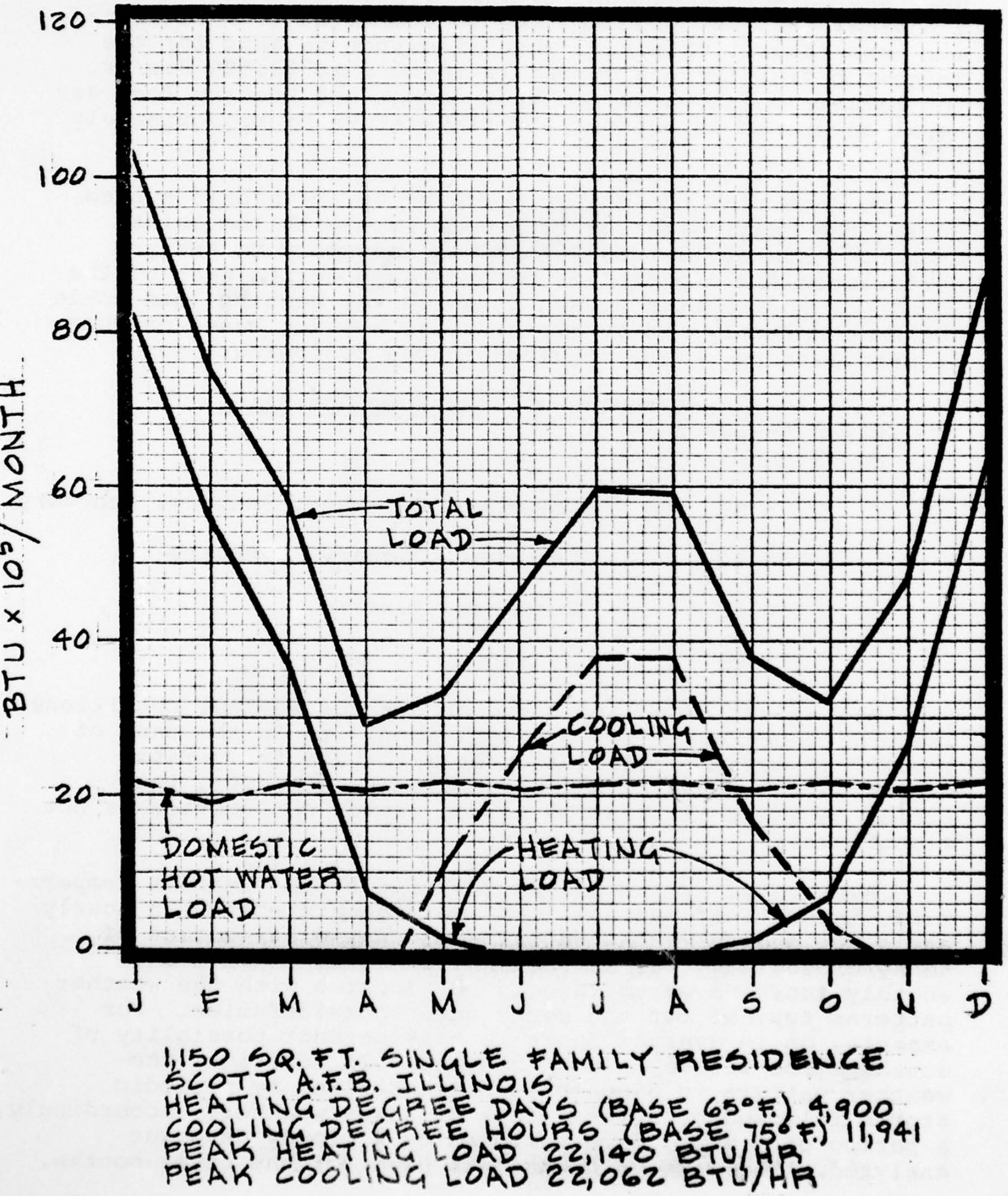


Figure 67. Annual Load Distribution - Scott AFB

based on the air-to-air heating performance and because both locations experience the same peak conditions, the same size unit was selected for each location. The unit selected has a nominal heating output of 38,000 Btu/hr at 40°F outdoor air temperature. However, when allowance is made for the defrost cycle, the net output is reduced to 33,500 Btu/hr. The balance point where net heat output and heating load are equal occurs at an outdoor air temperature of approximately 17°F.

To determine the energy consumed by the hybrid system during the heating season for comparison with the solar augmented system (System #6) it was necessary to analyze performance for a period of several days during each of the months of the heating season in which the heating load could not be completely met by direct solar heating based on Phase II calculations. This resulted in the following:

Collector Area (Based on average conditions)	Months of Heat Pump Operation
75 SF	November, December, January, February
100 SF	December, January, February
150 SF	December, January
200 SF	January

All calculations are based on average weather conditions. There will obviously be occasions when, (as in the case of the 200 SF system,) the heat pump will have to operate during December due to weather extremes. However, the effect on the total heating season energy consumption is not significant.

Average sunny and cloudy day conditions (ambient temperature, insolation, heating load) were determined on an hourly basis for each day. A typical seven day distribution of sunny/cloudy days was established based on the overall monthly sunshine percentage in conjunction with the weather patterns typical for the month under consideration. For example, on an average there is a 44 percent possiblity of sunshine for Little Rock in January. In addition, the weather pattern in January is characterized by periodic storms followed by several days of sunny weather. Accordingly, a period of 3 sunny days followed by 4 cloudy days was analyzed. The same procedure was used for the other months.

In general, the analysis was done for a 7-day period for each month but the actual controlling factor was the length of time required to achieve system equilibrium. In other words, for a typical period in any given month, there exists a state in which the relationship among storage temperature, solar energy collected, heat pump operation and heating load is such that the day-to-day patterns become repetitive. The analysis was extended until this point was reached.

The calculation form shown in Appendix III was used for the above analysis. Shifts in modes of operation were determined, based on the control system logic shown in Figure 60. Collector performance was based on a typical liquid collector efficiency curve. Heat pump performance was based on Figures 54 and 55. The energy consumed to meet the heating load for the period analyzed was summed and used as the basis for projecting energy consumption for the remainder of the month. This was then compared with the energy consumed by System #6, the solar augmented heat pump system.

Figure 68 shows the change in storage temperature over time for a typical 7-day period during January at Little Rock. It shows the relative effect of increasing collector area and indicates the number of cloudy days through which the solar system has some effect on the energy consumption in either the direct solar or water/air modes.

In order to test the sensitivity of the hybrid heat pump system to variations in storage volume, system performance was analyzed for a typical January period using 4 gal/SF of collector and 1.5 gal/SF. The results indicate that energy consumption would increase by approximately 10 percent for the larger storage and decrease slightly (less than 5 percent) for the smaller storage. Conclusions are:

a. Larger storage volumes are detrimental to system performance because it takes considerably longer for the system to reach direct utilization temperature. Thus, the heat pump operates in the water/air mode for many more hours than a system with less storage.

b. Small energy and cost savings may be possible by decreasing the storage volume slightly, however, significant changes in energy consumption are not likely.

Performance of the solar assisted hybrid heat pump system at Scott AFB was analyzed for January in the same manner as previously described for Little Rock. January was chosen for analysis because the maximum heating loads occur during that month. If the hybrid heat pump system does not

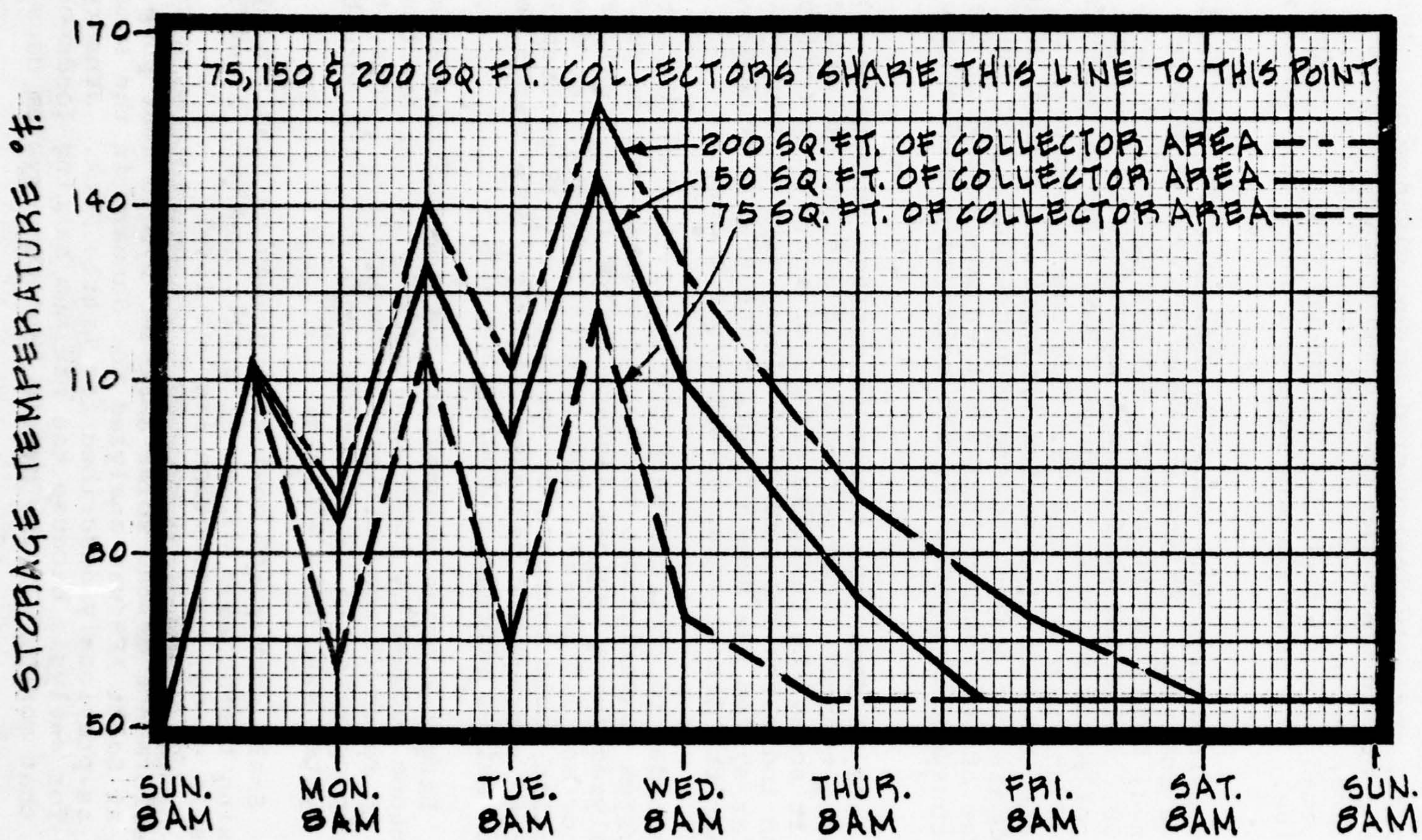


Figure 68. System No. 22 - 7 Day Temperature Profile

produce energy savings during January, it would be even less effective during the remainder of the heating season when either the loads are less, or periods of direct solar heating are longer, or both.

4.9 SYSTEM COMPARISON

The results of the performance analysis for the systems at Little Rock AFB are shown in Table 11. A comparison with System #6 performance for the three equal collector areas is shown below.

Collector Area (Sq Ft)	Annual KWH Saved System #6	Annual KWH Saved System #22	Annual Solar % Participation System #6	Annual Solar % Participation System #22
100	4731	4370	67	62
150	5396	5059	75	70
200	6087	5908	83	79

Table 12 shows a monthly comparison of energy required to meet the heating and DHW loads for collector areas of 100 and 150 SF.

The results indicate that the solar assisted hybrid heat pump system consumes about 5 percent more energy than the solar augmented heat pump system when meeting the same heating and DHW loads.

The results of the analysis for January at Scott AFB are shown in the following table.

Energy Required to Meet Heating and DHW Loads - KWH

Collector Area	System #6	System #22	% Increase
150 SF	1578	2001	27
200 SF	1442	1871	30
250 SF	1304	1760	35

The above results are considerably more significant than those obtained for Little Rock. The explanation for the increased energy consumption of the hybrid system is related to the relative lengths of time the system operates in the direct solar heating mode, the mode which consumes the least energy.

Table 11. System No. 22 Hybrid System Performance - Little Rock AFB

COLLECTOR AREA SQ. FT.	ANNUAL SOLAR PARTICIPATION %	ENERGY REQUIRED FOR HEATING KWH	ENERGY REQUIRED FOR DHW KWH	ENERGY REQUIRED TO OPERATE SOLAR SYSTEM KWH	TOTAL ANNUAL KWH CONSUMED	TOTAL ANNUAL ENERGY REQUIRED BY CONVENTIONAL SYSTEM - HEATING & DHW KWH	ANNUAL SAVINGS KWH	ANNUAL SAVINGS $
75	56	1,095	3,086	1,496	5,677	9,592	3,915	87
100	62	943	2,707	1,572	5,222	9,592	4,370	97
150	70	650	2,231	1,652	4,533	9,592	5,059	112
200	79	280	1,758	1,646	3,684	9,592	5,908	131

Table 12. Energy Required to Meet Heating and DHW Loads - Little Rock, AFB

MONTH	AUGMENTED SYSTEM KW	HYBRID SYSTEM KW	AUGMENTED SYSTEM KW	HYBRID SYSTEM KW
Jan.	1,141	1,268	1,005	1,181
Feb.	701	835	691	691
March	439	439	326	326
April	131	131	138	138
May	131	131	138	138
June	131	131	138	138
July	131	131	138	138
Aug.	131	131	138	138
Sept.	131	131	138	138
Oct.	131	131	138	138
Nov.	670	670	343	343
Dec.	993	1,093	865	1,026
TOTAL	4,861	5,222	4,196	4,533

The hybrid system draws the storage temperature down to 55°F in the water-to-air mode of operation. The solar augmented system only draws the storage down to 100°F at which time it shifts to the air-to-air mode. When the collection of solar energy resumes, the augmented system is able to operate in the direct solar mode immediately while the hybrid system operates in the water-to-air or air-to-air mode until storage reaches 100°F.

To show the above relationship graphically, the day-long performance of the solar augmented system was analyzed for a typical sunny day in January and compared with the performance of the hybrid system. Figure 69 shows the results for Little Rock on an hourly basis. The solar augmented system begins the day with a storage temperature of 100°F compared to 75°F for the hybrid system. During the collection period, the storage temperatures reach 146°F in the augmented system and 129°F in the hybrid system. Consequently, the augmented system is able to provide direct solar heating until approximately 5 a.m. the following morning at which time it changes to the air-to-air mode of operation. The hybrid system is only able to provide direct solar heating until approximately 1 a.m. at which time it changes to the water-to-air mode of operation. Figure 70 shows the results of the day-long system operation described above on a cumulative power input basis. The cumulative difference at the end of the day is approximately 1900 watts.

The same analysis was performed for Scott AFB with the results shown in Figures 71 and 72. The difference in energy consumption is considerably greater than at Little Rock. The solar augmented system begins the day at 100°F while the hybrid system begins at 70°F. The hybrid system operates in the water-to-air mode until it reaches direct utilization temperature at approximately 1 p.m. and reaches a maximum temperature of 124°F compared to 145°F for the augmented system.

The augmented system provides direct solar heating until approximately 5 a.m. while the hybrid system must shift to water-to-air at approximately 11 p.m. The cumulative consumption in Figure 72 indicates a day-long difference of over 6000 watts.

The preceding analysis of system performance in locations with differing climatic conditions provides two data points on a curve relating heating degree days to energy consumed for heating. Because the detailed analysis for Scott AFB was for January only, Figure 73 shows the relationship between the energy required to meet the January heating load and degree days for the hybrid and augmented systems.

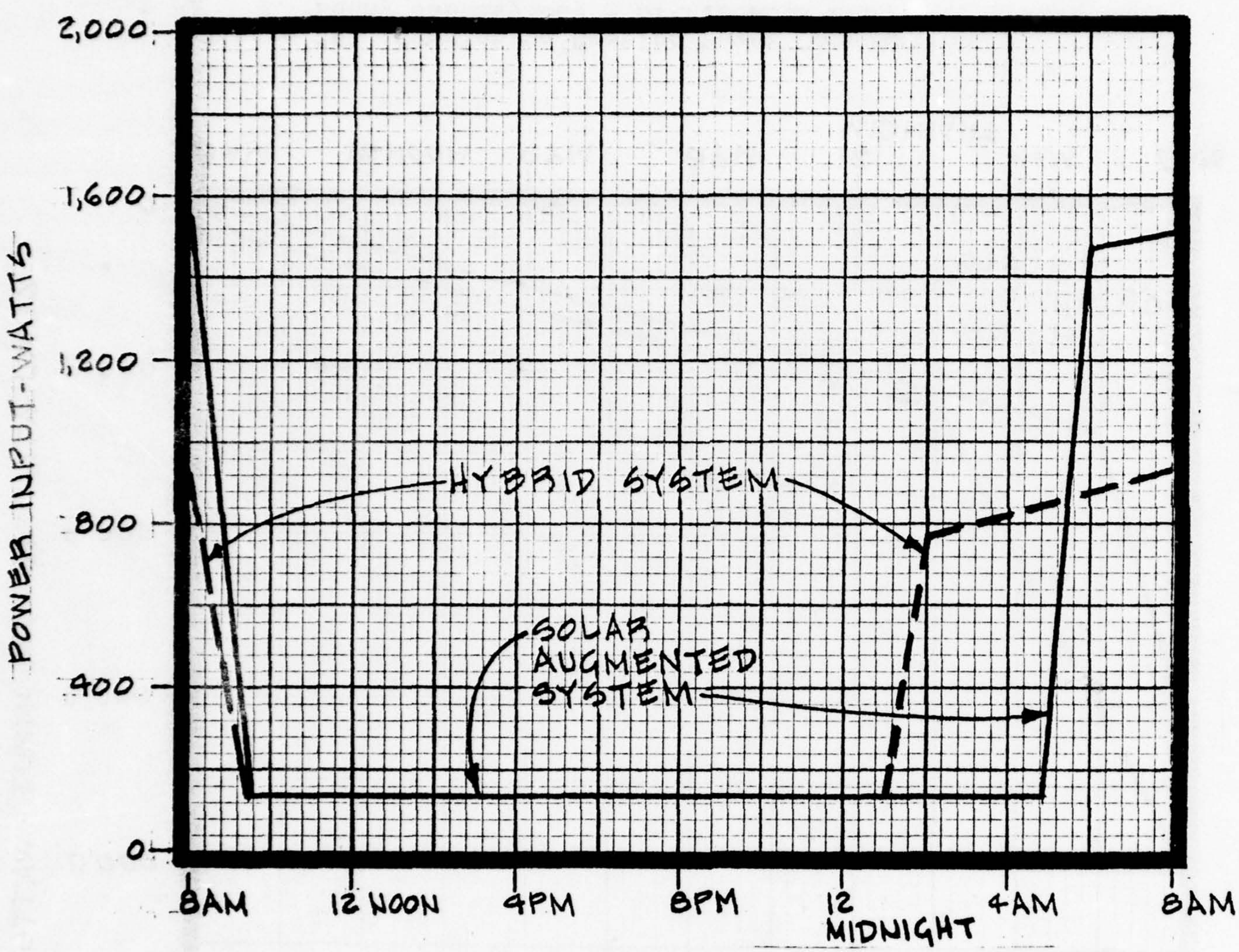


Figure 69. Heating Power Consumption - Sunny January Day
Little Rock AFB - 100 Sq Ft Collector Area

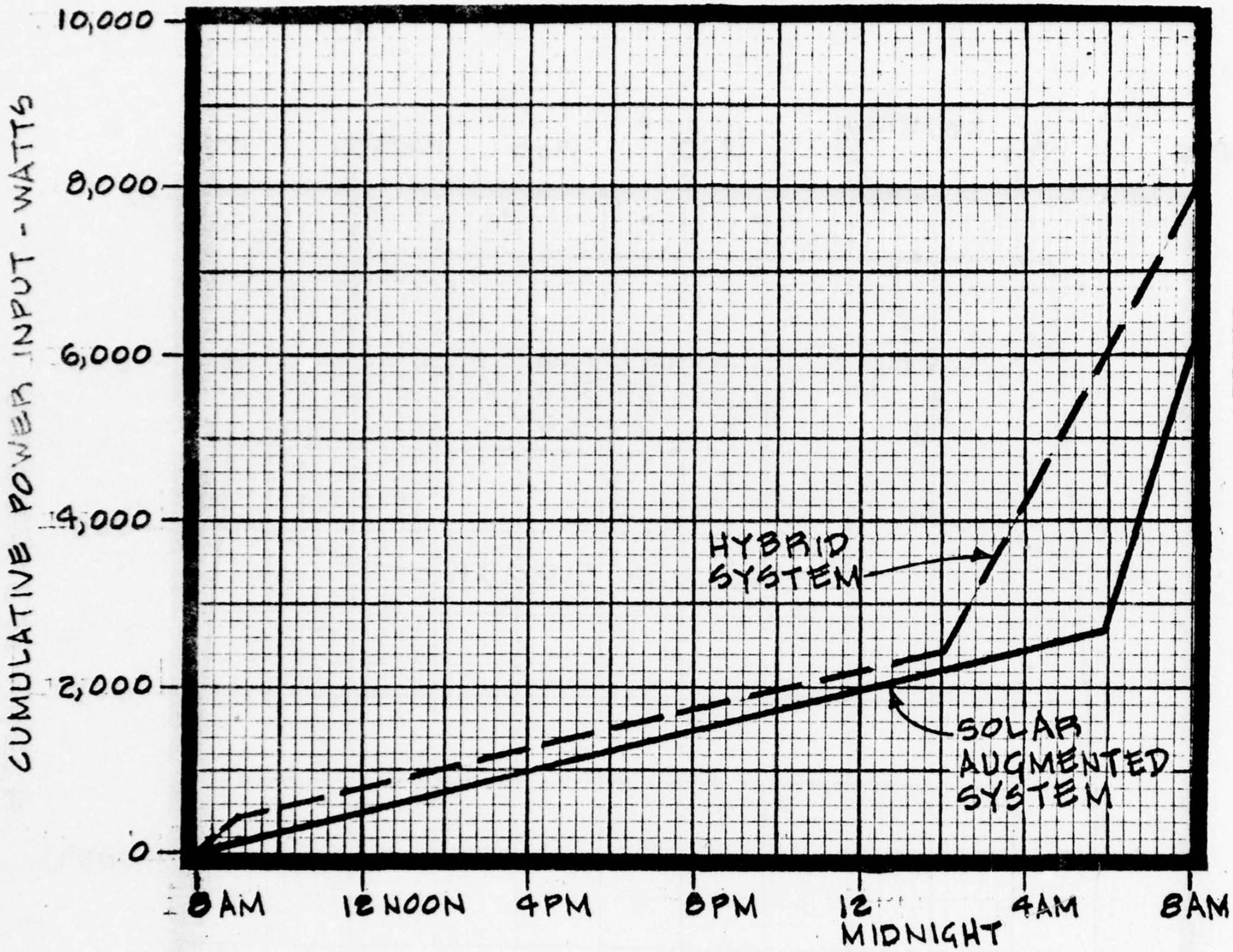


Figure 70. Cumulative Heating Power Consumption
Sunny January Day - Little Rock AFB
100 Sq Ft Collector Area

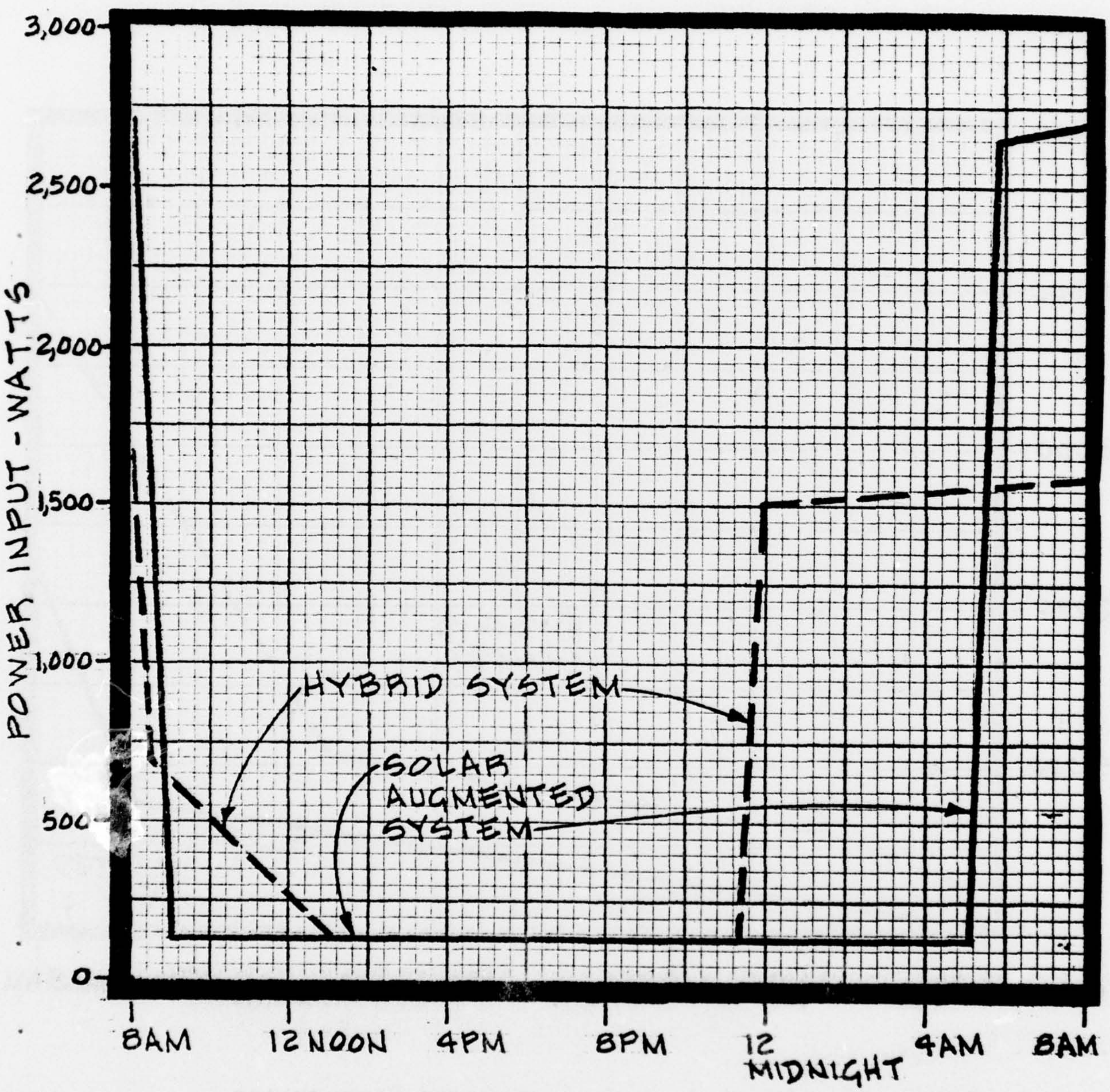


Figure 71. Heating Power Consumption
Sunny January Day Scott AFB
200 Sq Ft Collector Area

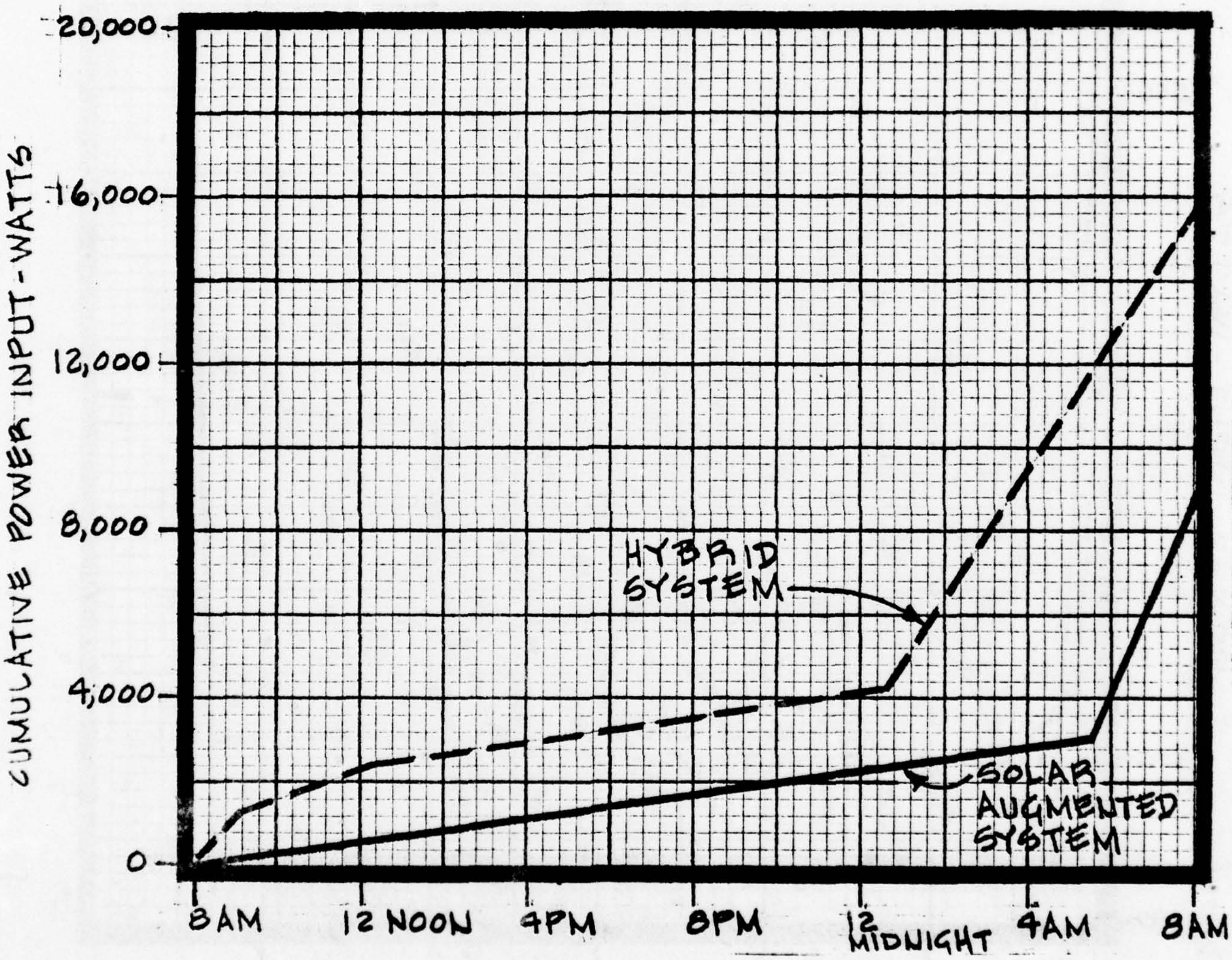


Figure 72. Cumulative Heating Power Consumption
Sunny January Day - Scott AFB
200 Sq Ft Collector Area

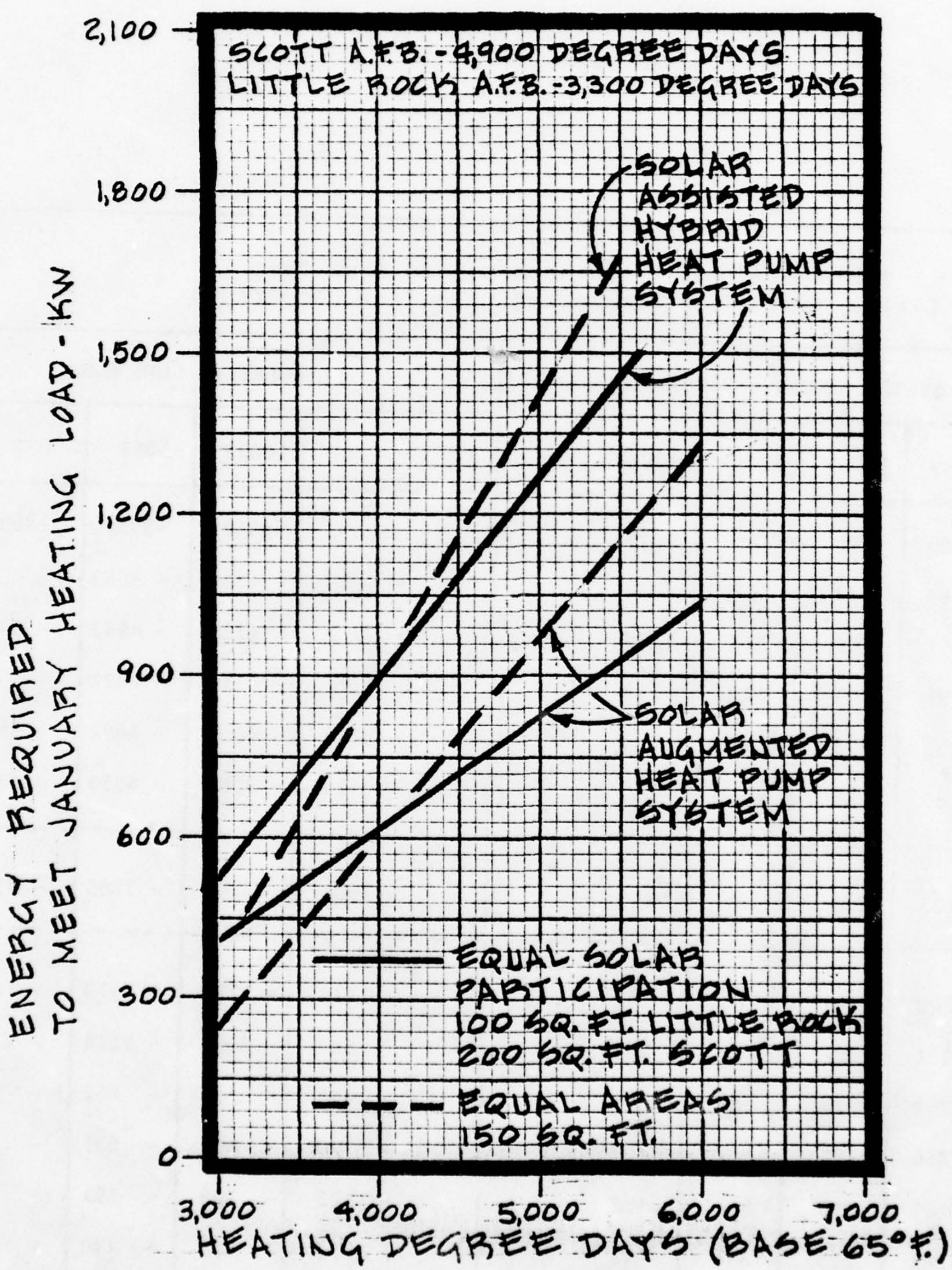


Figure 73. System Performance VS Degree Days

Table 13. Solar Assisted Hybrid Heat Pump Little Rock AFB - Cumulative Cash Flow Comparison								
SOLAR SYSTEM ANNUAL SAVINGS					CUMULATIVE CASH FLOW			
YEAR	75SF	100SF	150SF	200SF	75SF	100SF	150SF	200SF
0	$0	$0	$0	$0	-$3995	-$4425	-$5175	-$6000
1	87	97	112	131	- 3908	- 4328	- 5063	- 5869
2	91	102	118	138	- 3817	- 4226	- 4945	- 5731
3	96	107	123	144	- 3721	- 4119	- 4822	- 5587
4	101	112	130	152	- 3620	- 4007	- 4692	- 5435
5	106	118	136	159	- 3514	- 3889	- 4556	- 5276
10	135	151	174	203	- 2901	- 3205	- 3766	- 4352
20	220	245	283	331	- 1118	- 1218	- 1472	- 1668
21	231	257	298	348	- 887	- 960	- 1174	- 1321
22	242	270	312	365	- 645	- 690	- 862	- 956
23	254	284	327	383	- 390	- 406	- 535	- 573
24	267	298	344	402	- 123	- 108	- 191	- 170
25	281	313	361	422	+ 157	+ 205	+ 170	+ 252

Table 14. Cumulative Cash Flow Comparison
Little Rock AFB

	SOLAR SYSTEM ANNUAL SAVINGS				CUMULATIVE CASH FLOW			
	100SF		150SF		100SF		150SF	
YEAR	HYBRID	SOLAR AUGMENTED	HYBRID	SOLAR AUGMENTED	HYBRID	SOLAR AUGMENTED	HYBRID	SOLAR AUGMENTED
0	$0	$0	$0	$0	-$4425	-$4005	-$5175	-$4755
1	97	105	112	125	- 4328	- 3900	- 5063	- 4630
2	102	110	118	131	- 4226	- 3790	- 4945	- 4499
3	107	116	123	138	- 4119	- 3674	- 4822	- 4361
4	112	122	130	145	- 4007	- 3552	- 4692	- 4216
5	118	128	136	152	- 3889	- 3424	- 4556	- 4064
10	151	163	174	194	- 3205	- 2683	- 3766	- 3181
20	245	265	283	316	- 1218	- 533	- 1472	- 619
21	257	279	298	332	- 960	- 252	- 1174	- 287
22	270	293	312	348	- 690	+ 41	- 862	+ 61
23	284		327		- 406		- 535	
24	298		344		- 108		- 191	
25	313		361		+ 205		+ 170	

Curves for equal areas and equal solar participation are shown. The result clearly indicates that the hybrid system consumes more energy than the augmented system in each location. However, of even greater significance is the fact that the curves are diverging with increasing degree days indicating that the hybrid system does not become more effective in colder climates. However, it should be pointed out that Figure 73 is based on two locations only. Analysis of several locations over a wide range of degree days would give more definitive results.

The cash flow analysis is based on the same assumptions made in the Section III of this report--a net increase in energy cost of 5 percent per year after inflation and a present energy cost of $0.0222 per KWH. Table 13 shows a cumulative cash flow comparison between the four sizes of hybrid heat pump systems analyzed at Little Rock and indicates no significant differences. Table 14 shows a cumulative cash flow comparison between the hybrid system and the augmented system for collector areas of 100 and 150 SF. The results indicate that the solar augmented system is economically more attractive.

SECTION V

PHASE III

5.1 METHODOLOGY

The general approach was to integrate the System No. 6 design presented in Phase II into a building at Little Rock AFB. This began with a site visit conducted on August 5, 1977. The purpose of the visit was to select a specific building, to determine existing conditions within the building, and to investigate the feasibility of various energy conservation improvements. Building load calculations were refined for use in selecting equipment of proper capacity and projecting energy consumption. Collector array area was optimized based on cost and savings.

A drawing was prepared showing the final flow diagram, control diagram, and plan/elevation views of the building. Collector array configuration, equipment location and piping layout are also shown. A preliminary specification of the major equipment was written.

Equipment and installation costs were estimated and, in conjunction with projected savings, the approximate payback period was calculated.

5.2 SITE VISIT

The site visit was made to accomplish three specific tasks:

1. Evaluate the existing conditions as they affect system design and installation.

2. Determine the feasibility of various energy conservation measures.

3. Select a specific building for the solar augmented heat pump system installation.

The original heat pumps installed in 1957 were manufactured by the Mathes Company which has subsequently ceased operations. As failures have occurred, base maintenance personnel have replaced the original components so that the units today bear little resemblance to those originally installed. The average coefficient of performance of the existing units is not known.

The existing refrigerant piping between the indoor and outdoor units runs beneath the house in a pipe which, according to maintenance personnel, is frequently filled with water contributing to a loss in efficiency. They requested that the refrigerant lines between the new units be routed up the exterior wall, through the attic and down to the in-door unit.

New thermostats which limit the minimum cooling season and maximum heating season settings to 75°F are being installed along with new outdoor air temperature sensors which prevent the supplementary electric heaters from operating above certain outdoor temperature. The solar system controls will be integrated with the existing temperature control system. Radio controllers have been installed on some of the heat pump units to permit remote shutdown for short time periods during the cooling season. The unit at 170 Alabama has such a unit which will have to be connected to the new heat pump.

An ongoing hot water heater installation program has been implemented to replace the existing heaters as failures occur. Tank capacity is 40 gallons with a 4500 watt electric element.

There is an active energy conservation program underway at Little Rock Air Force Base. Additional fiberglass insulation has been blown into the attic reducing the overall "U" value to 0.033. The housing units presently have single hung aluminum frame windows. A contract, to be completed by 1978, for the installation of storm windows and insulated doors is out to bid. A re-siding program which includes additional insulation on the exterior of the building is also underway. The overall "U" value of the walls will be reduced to 0.055 after siding installation. Electric resistance heaters presently installed in each bath are being removed and replaced by branch ducts supplying 100 cfm each. This will provide two spare 20 amp 220 volt circuit breakers.

The type "E" residence was used as the basis of design for Phase II, because several had favorable orientation. Also, in selecting the residence two other conditions had to be met (1) the collector array had to face the rear of the house and (2) it should have been retrofitted or included in the retrofit projects currently being accomplished. The base asked that the collector array be installed on the rear of the residence to reduce the visibility of the project. It is anticipated that the installation will arouse the curiosity of the community and be a source of annoyance to the residence. With these limitations, along with trying to keep the orientation as close to true south as possible the choice was narrowed down to (1) 170 Alabama, or (2) 179 Illinois. We have assumed that the system will be installed at 170 Alabama.

A site plan illustrating orientation and shading from backyard trees is shown in Figure 74. Because the trees closest to the house are deciduous and the shading angle is very low, wintertime shading will be insignificant.

The equipment required for the solar system must be located in a new enclosed and insulated space behind the carport. While the solar system is not operating during cold weather, a small electric heater set to maintain a minimum of 40°F will be required. In addition, it was requested that the air volume handled by the new indoor unit be held close to the present rate to prevent noise.

It was also requested that sufficient instrumentation be incorporated in the design to determine the energy consumed by the residence with the solar system.

5.3 ENERGY CONSERVATION

The building heating and cooling loads used in Phase II were based on the assumption that the thermal characteristics of the house would be improved by providing additional wall and roof insulation and double glazed windows. As discussed in the previous section an energy conservation program implementing the measures which were recommended in Phase II was already underway. The 'U' values of the building skin will be reduced to the recommended values. Additional insulation or window modifications would not be economical in the Little Rock area climate. Accordingly the load calculations are based on the building characteristics which will exist after the energy conservation improvements currently underway have been completed. These heating/cooling load characteristics are included in Appendix IV.

5.4 SYSTEM CHARACTERISTICS

The operating characteristics of the system are essentially the same as described in Phase II. Some minor modifications were made to the flow diagram and are in-corporated in the final flow diagram shown in Figure 75.

The following description of System 6 is identical to that given in the Phase II section of this report, but is included here for ease of reference.

The system employs liquid collectors and water storage. Frost protection is achieved by automatic drain down.

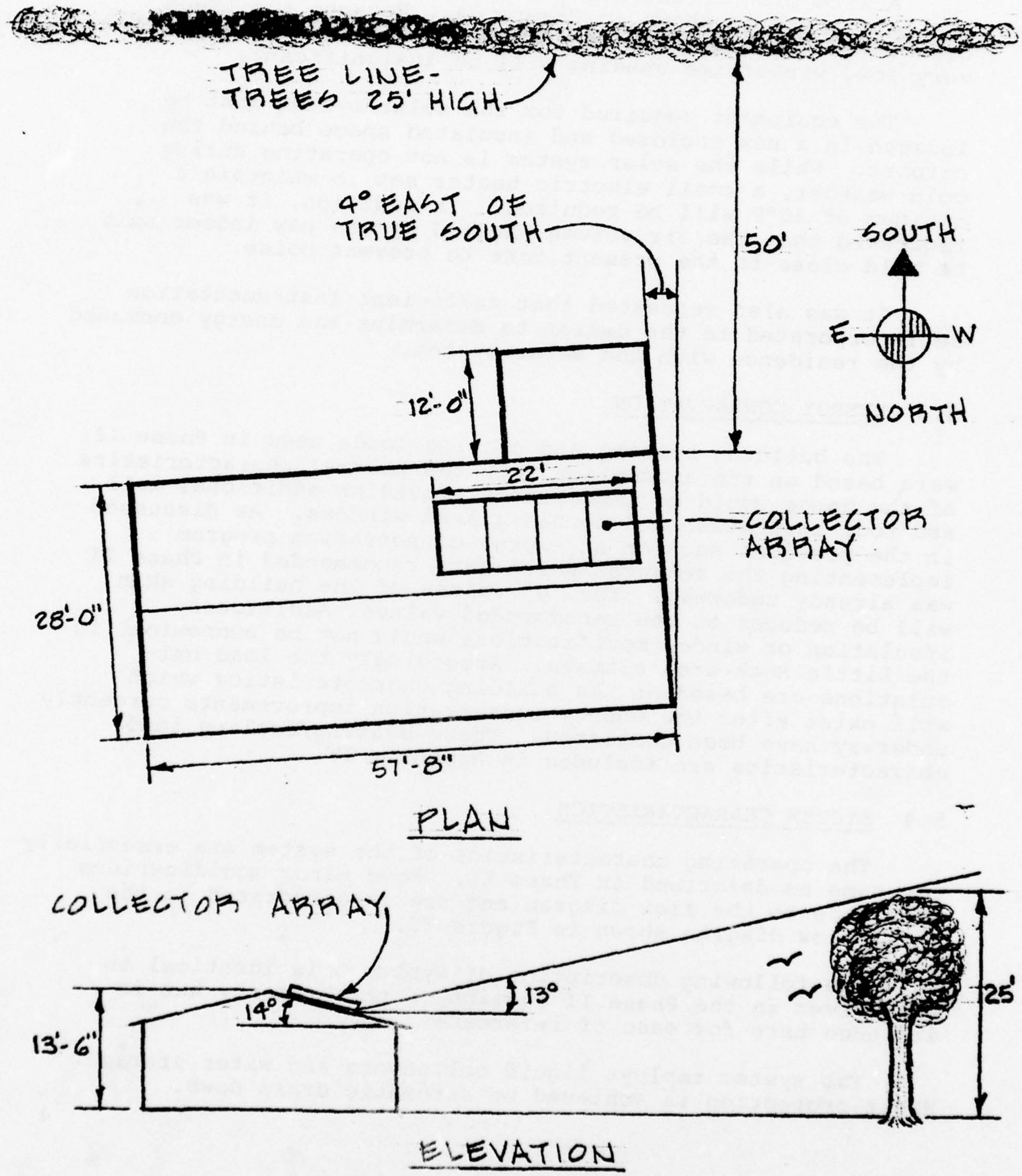


Figure 74. Orientation/Shading #170 Alabama

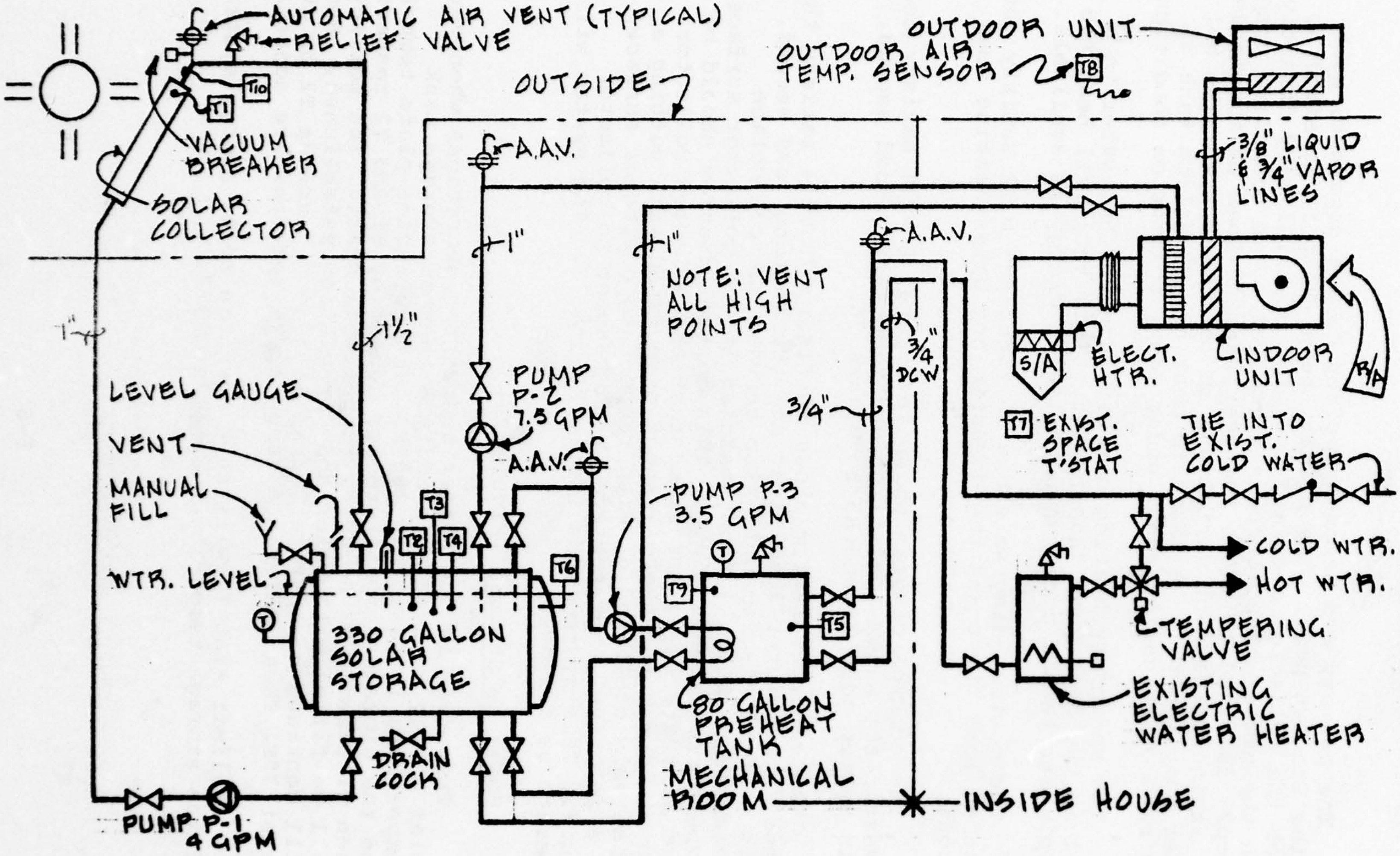


Figure 75. System Flow Diagram

The primary and secondary circuits are separate, each having its own pump. The primary pump (Pump 1) is sized to match the collector area and its capacity will vary directly with the number of collectors installed. The secondary pump (Pump2) is sized to match the peak heating load of the house and is independent of variations in collector area. The hot water pump (Pump 3) is sized to heat the 80 gallon tank at the rate of 20 gallons/hour under ideal collection conditions.

The secondary heating circuit supplies hot water to a coil in the heat pump indoor unit. This coil will heat the supply air to the house whenever solar energy is available.

The control diagram shown in Figure 76 will satisfy the heating load first and the domestic hot water heating load second.

The control diagram shown in Figure 77 will satisfy the domestic hot water load first and the heating load second. This system adds T9 and R3 to Figure 76.

The control diagram shown in Figure 78 will satisfy the domestic hot water load first and the heating load second. This control diagram will also prevent the collection system pump (Pump 1) from starting if the collector surface temperature exceeds 250°F. This control diagram should be used if there is no assurance that the selected collector can withstand the thermal shock associated with putting a relatively cool liquid, with respect to collector surface, on a relatively hot surface, with respect to collector fluid. This system adds T10 to Figure 77. The system will operate as follows:

<u>Solar to Storage</u> (Figure 79)

This mode is for summer and winter operations whenever solar energy is available and the thermal storage tank temperature is less than 200°F. The collector plate temperature and the storage temperature are sensed by T1 and T2 respectively. When T1 exceeds T2 by 20°F, Pump 1 will start. The collectors will be filled with water and circulation established. Pump will continue to operate for as long as T1 exceeds T2 by at least 3°F. These reference temperature settings are adjustable.

A limit stat T2 will shut down the solar energy system if the storage temperature exceeds 200°F.

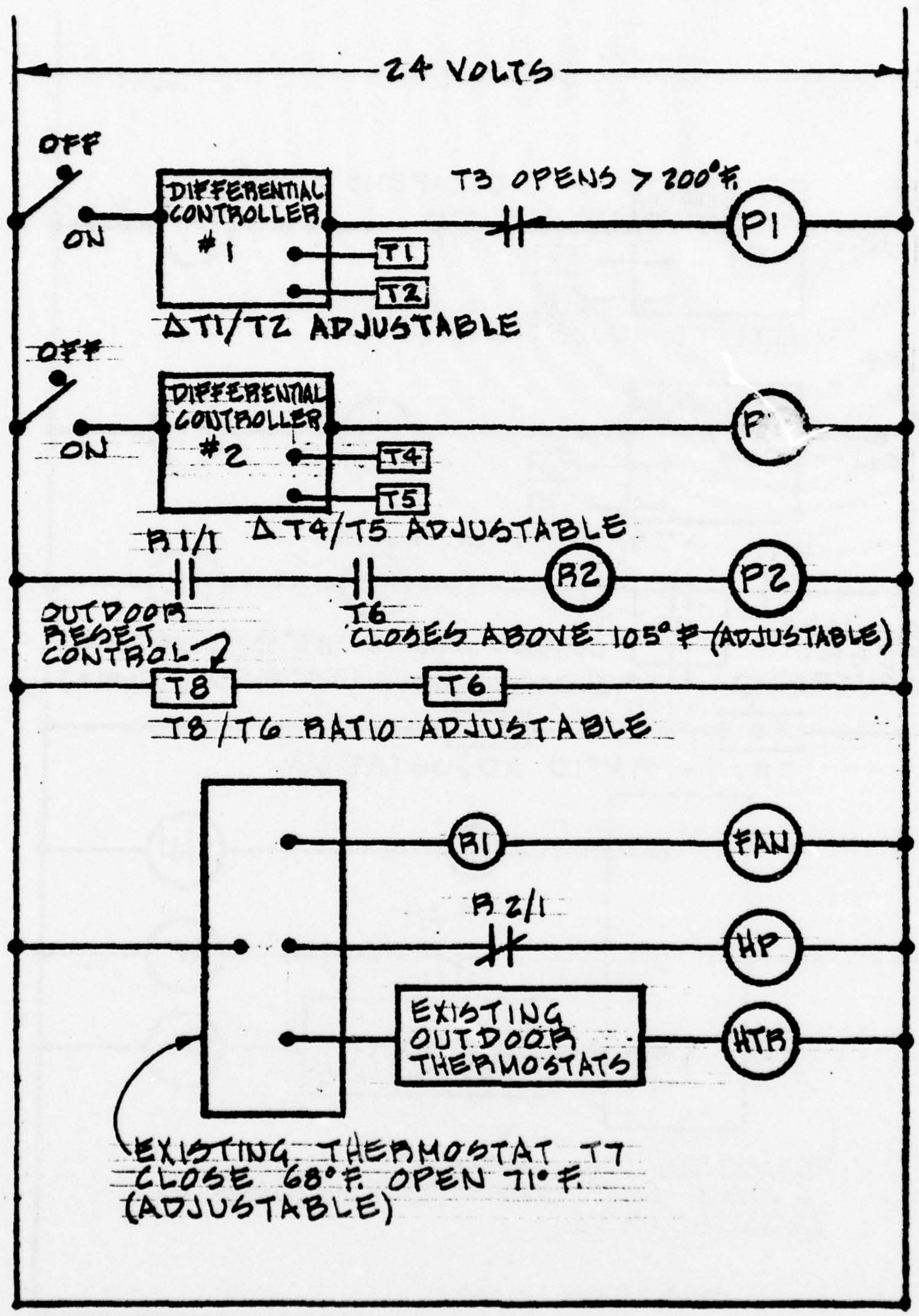


Figure 76. Control Diagram - Heating Load Satisfied First

THIS PAGE IS BEST QUALITY PRACTICABLE
FROM COPY FURNISHED TO DDC

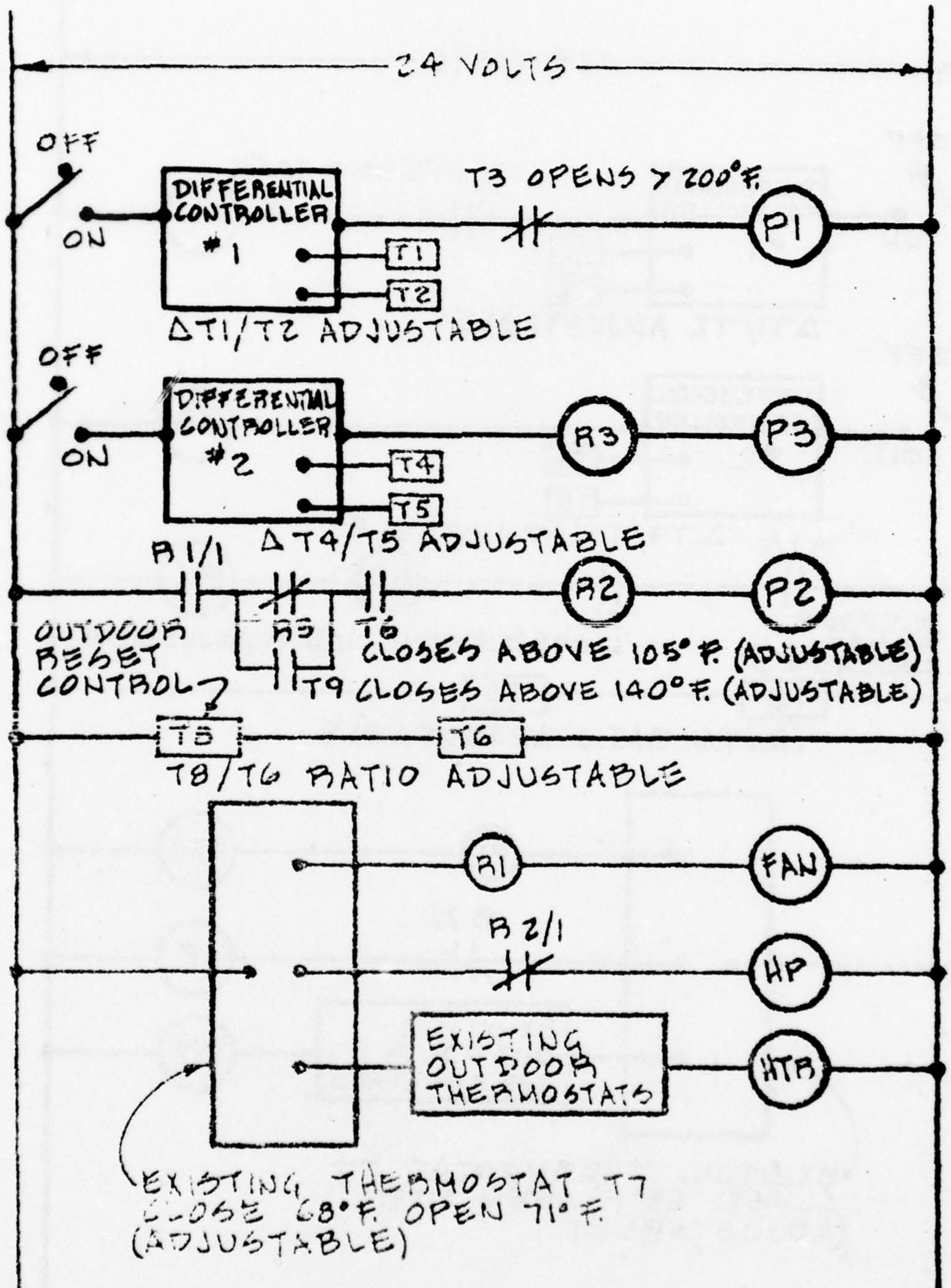


Figure 77. Control Diagram - DHW Load Satisfied First

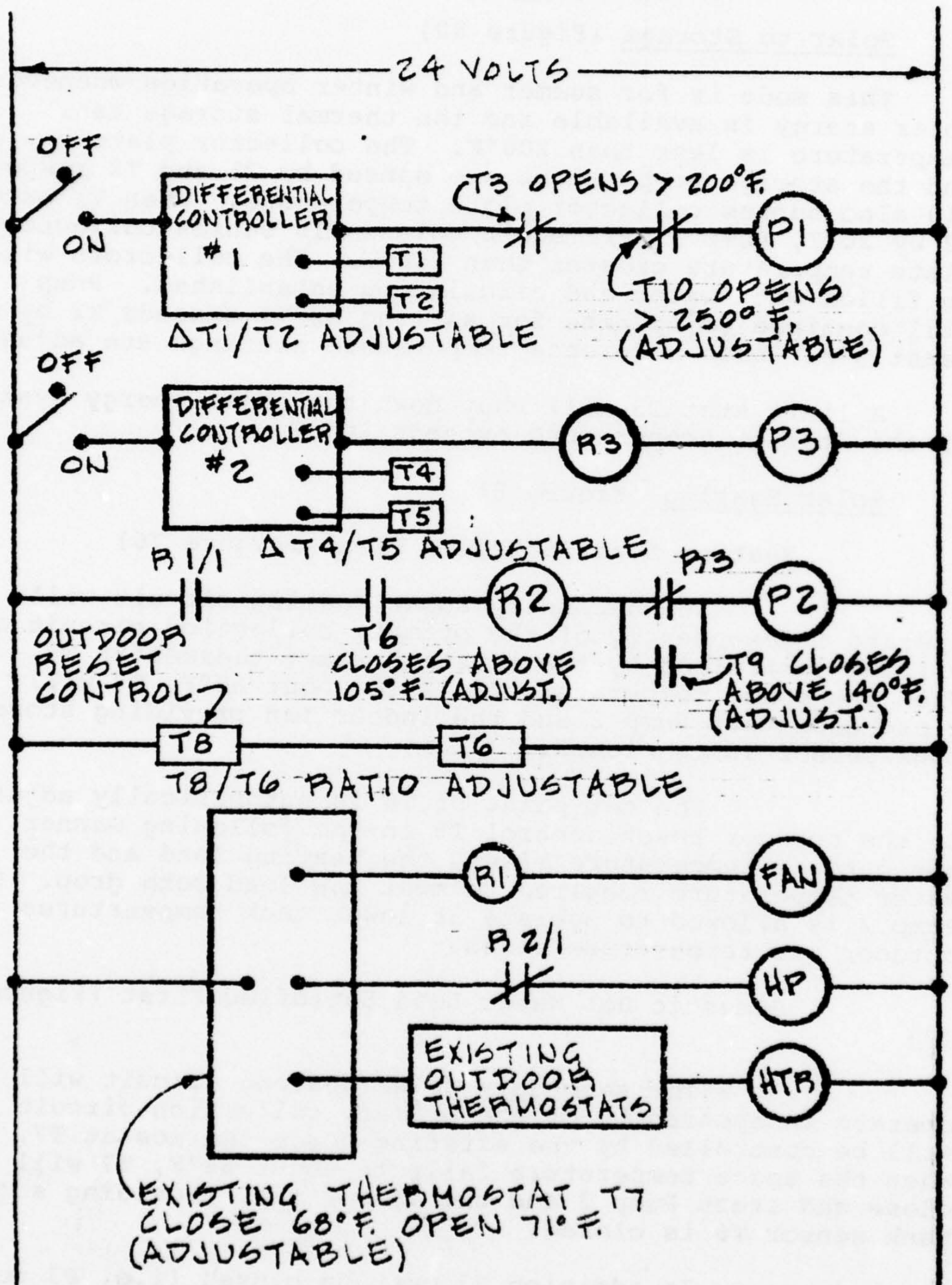


Figure 78. Control Diagram - With Collector Protection

Solar to Storage (Figure 80)

This mode is for summer and winter operation whenever solar energy is available and the thermal storage tank temperature is less than 200°F. The collector plate temperature and the storage temperature are sensed by T1 and T2 respectively. T10 also senses collector plate temperature. When T1 exceeds T2 by 20°F, Pump 1 will start unless T10 senses collector plate temperature greater than 250°F. The collectors will be filled with water and circulation established. Pump 1 will continue to operate for as long as T1 exceeds T2 by at least 3°F. These reference temperature settings are adjustable.

A limit stat T3 will shut down the solar energy system if the storage temperature exceeds 200°F.

Solar Heating (Figure 81)

Heating Load Satisfied First (Figure 76)

The secondary solar heating circuit will operate independently of the primary collection circuit and will be controlled by the existing space thermostat T7. When the space temperature falls to about 68°F, T7 will close and start Pump 2 and the indoor fan providing storage tank sensor T6 is closed.

The set point of T6 is automatically adjusted by the outdoor reset control T8 in the following manner. As the outdoor temperature rises, the heating load and the water temperature required to meet the load both drop. Thus Pump 2 is allowed to operate at lower tank temperatures as outdoor air temperature rises.

Domestic Hot Water Load Satisfied First (Figure 77)

The secondary solar heating circuit will operate independently of the primary collection circuit and will be controlled by the existing space thermostat T7. When the space temperature falls to about 68°F, T7 will close and start Pump 2 and the indoor fan, providing storage tank sensor T6 is closed.

In addition R3 must be closed (i.e. P3 not running) or T9 must be closed (DHW Preheat Storage above 140°F). This is to prevent the solar heating cycle from operating unless the domestic hot water load is satisfied.

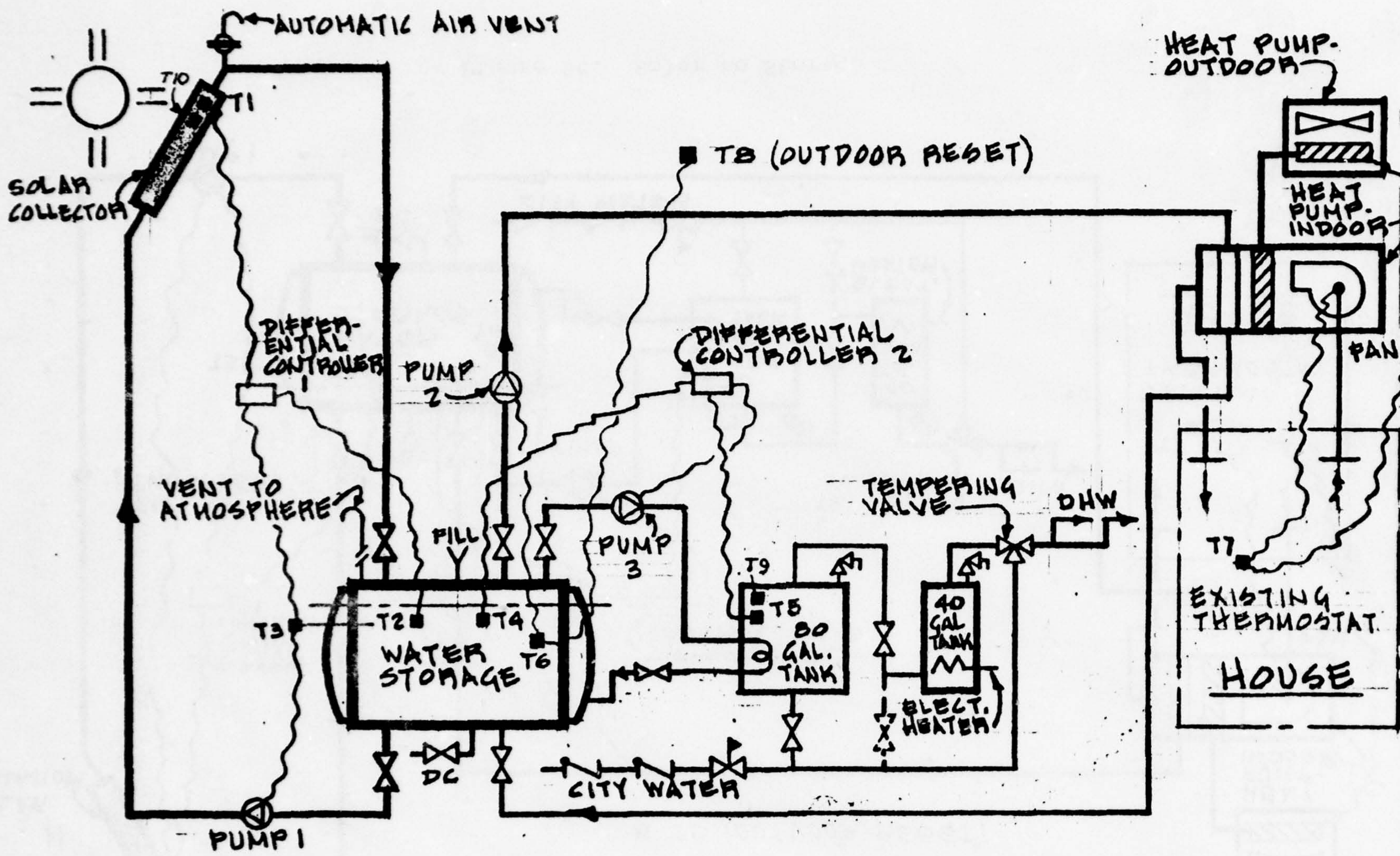


Figure 79. Solar to Storage

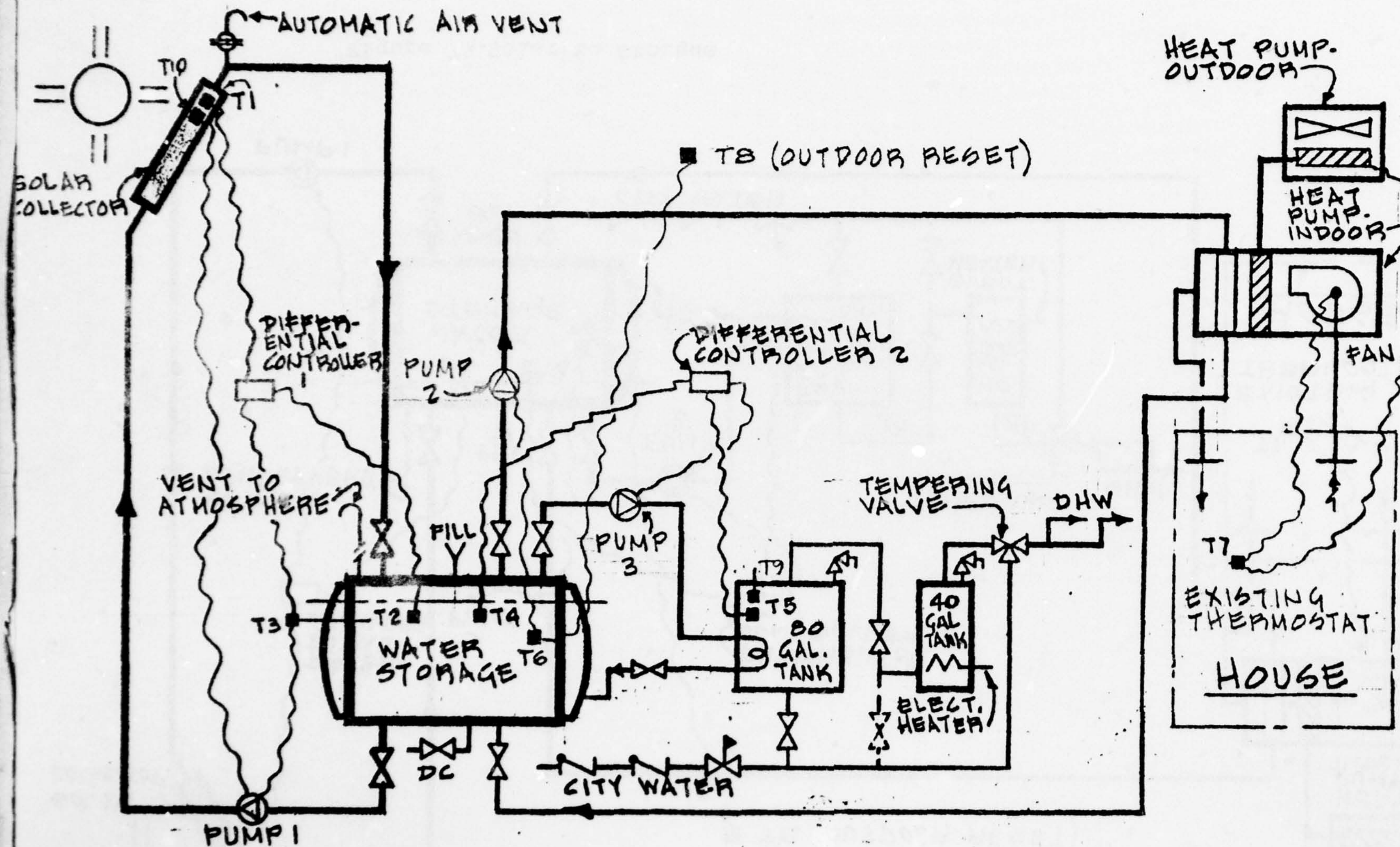


Figure 80. Solar to Storage

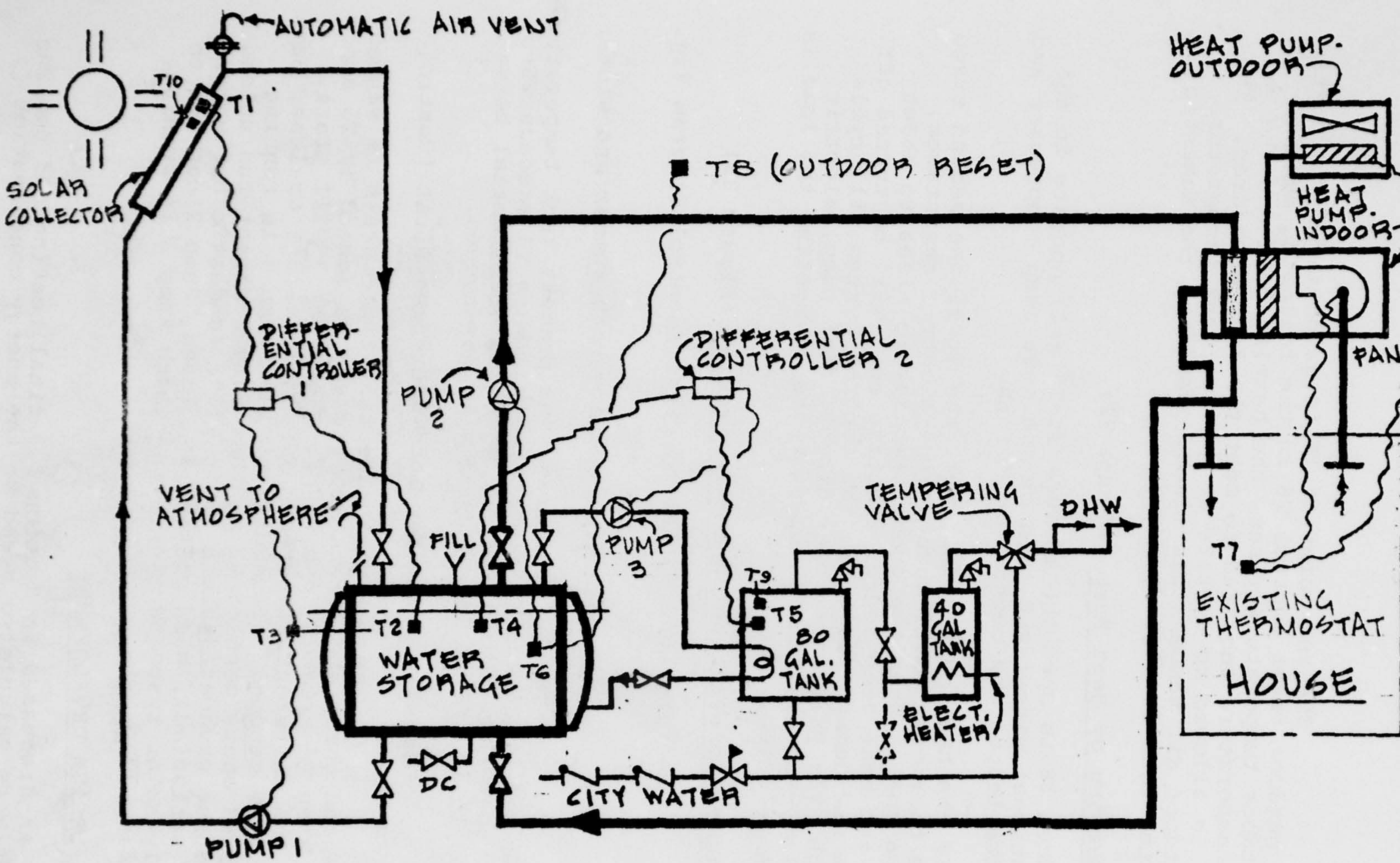


Figure 81. Solar Heating Direct

The set point of T6 is automatically adjusted by the outdoor reset control T8 in the following manner. As the outdoor temperature rises, the heating load and the water temperature required to meet the load both drop. Thus Pump 2 is allowed to operate at a lower tank temperature as the outdoor air temperature rises, providing the domestic hot water load is already met.

Heating by Heat Pump (Figure 82)

When T6 is open the thermostat T7 will operate in the conventional manner and turn on the heat pump compressor and both the indoor and outdoor unit fans.

If the heat pump cannot meet the load, the second stage of the thermostat will turn on the electric resistance heating coil. When the space temperature rises to about 71°F, the heat pump and/or electric coil will be turned off. Upon a new demand for heat, the control system will cycle through the priority--solar direct, heat pump, electric auxiliary--so that the cheapest means of meeting the load is always selected.

Solar Domestic Hot Water Generation (Figure 83)

Heating Load Satisfied First (Control Diagram Fig. 76)

This mode will operate both summer and winter.

The solar storage and preheat tank temperatures are sensed by T4 and T5 respectively. When T4 exceeds T5 by 10°F, then Pump 3 will start. When the differential between T4 and T5 is 3°F or less Pump 3 will be stopped.

Domestic Hot Water Load Satisfied First (Control Diagram Fig 77)

The solar storage tank temperature is sensed by T4 and the preheat tank temperature is sensed by T5 and T9. When T4 exceeds T5 by 10°F, then Pump 3 will start. When the differential between T4 and T5 is 3°F or less, Pump 3 will be stopped. In addition, when Pump 3 is running, relay R3 opens contact R3 so that Pump 2 cannot run unless T9 senses preheat storage above 140°F (domestic hot water load satisfied). When contact T9 closes, Pump 2 can run to supply solar space heating even though Pump 3 is running. See Fig. 77.

5.5 DESIGN DEVELOPMENT

As discussed in Section 5.3 final heating and cooling loads were calculated based on the energy conservation

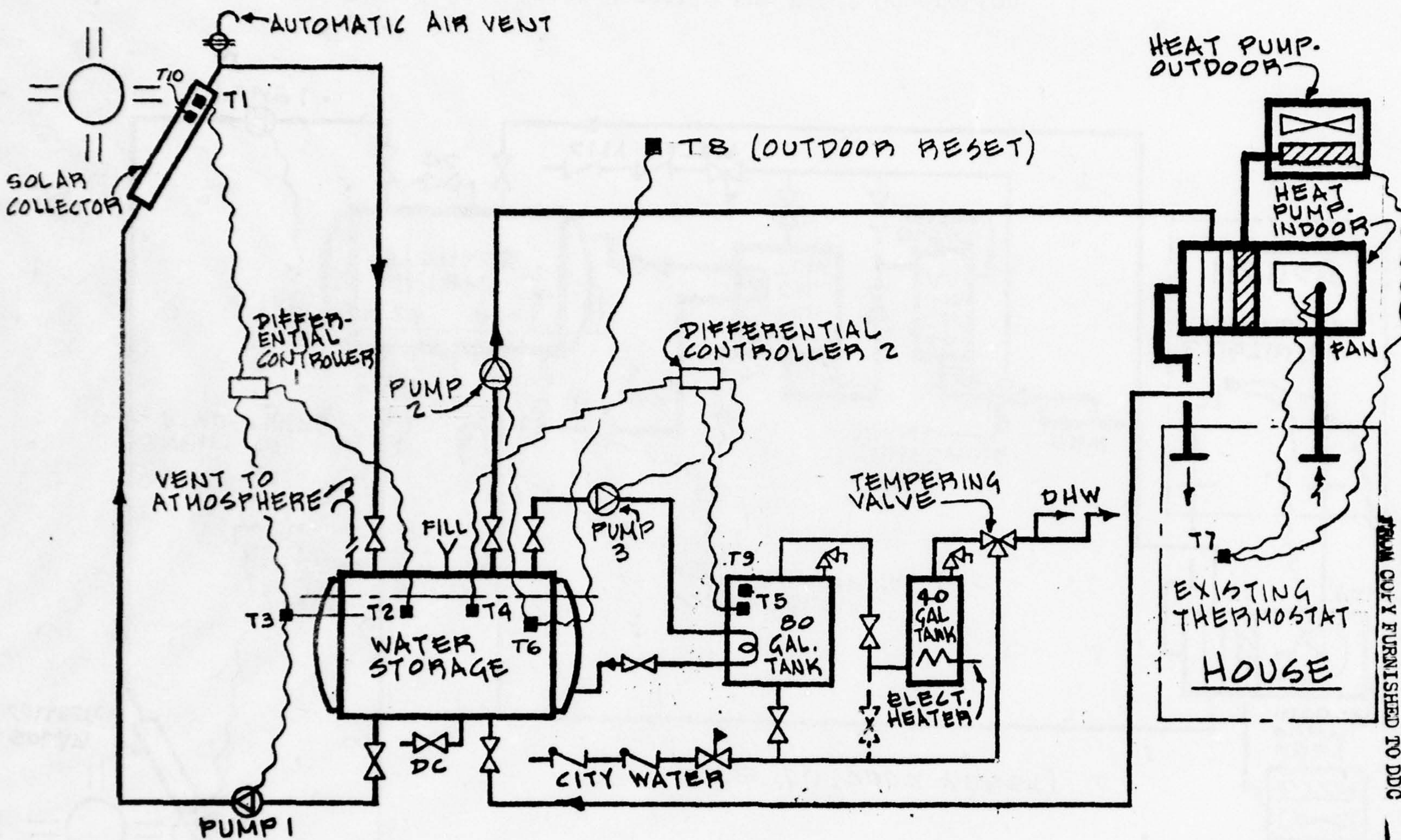


Figure 82. Heating by Heat Pump

THIS PAGE IS BEST QUALITY PRACTICABLE FROM COPY FURNISHED TO DDC

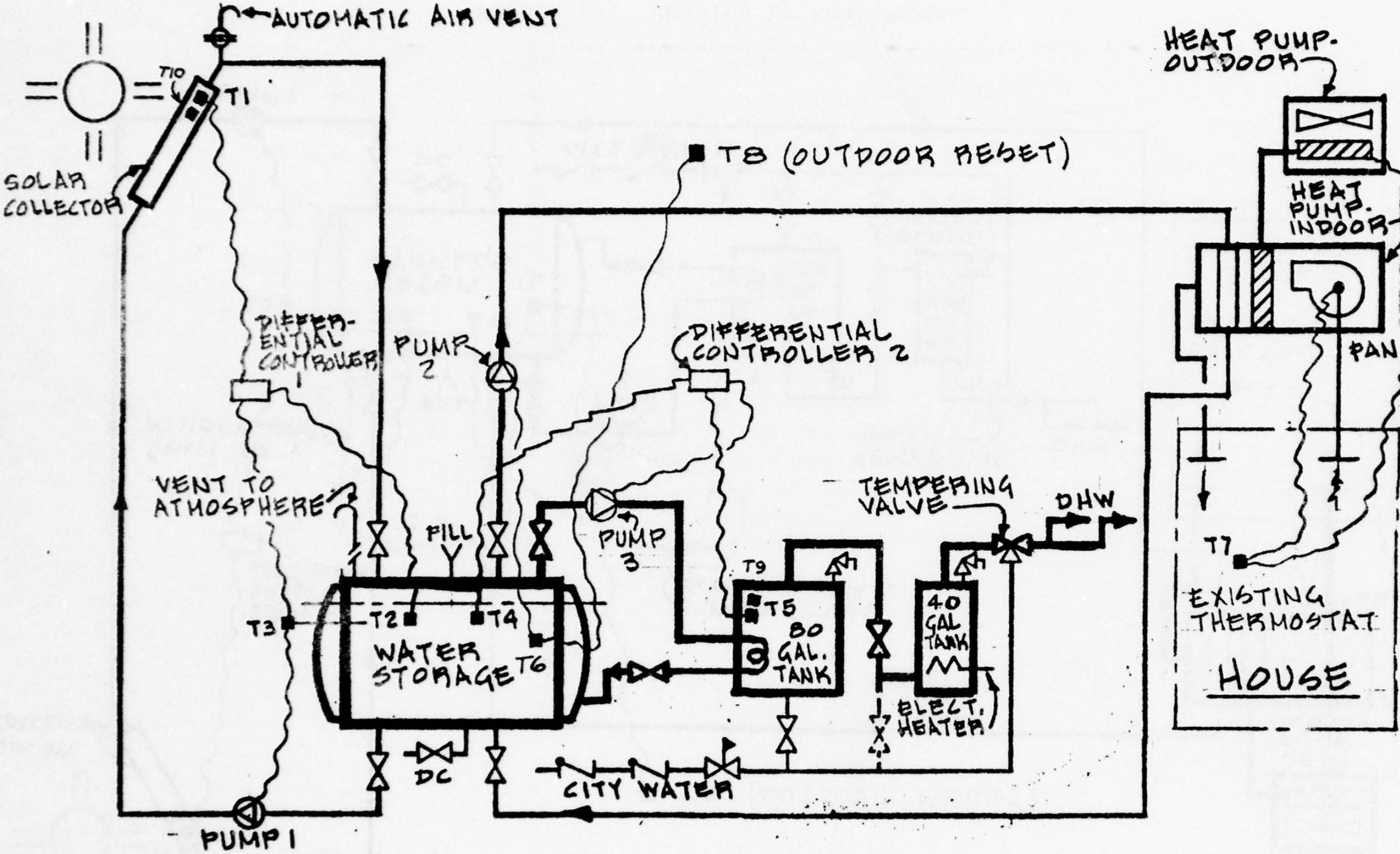


Figure 83. Solar Domestic Hot Water Generation

measures presently being carried out. The annual load distribution is shown in Figure 84. Heat Pump and solar collector sizing were based on these loads.

Heat pump selection was based on a computer analysis performed by Lennox Industries Inc. The analysis determines the amount of energy required to meet the heating load for units of various sizes. The two best selections were a 2 1/2 ton unit and a 3 ton unit. Although the 3 ton unit consumes slightly less energy on an annual basis, it will not provide adequate humidity control during the cooling season. This could result in uncomfortable conditions in the Little Rock climate. Accordingly, a 2 1/2 ton unit was selected for this system. Figure 85 which shows the heat output of the heat pump and the heating load over a range of outdoor air temperatures indicates a balance point of approximately 24°F. Since the Little Rock climate has an average of only 56 hours per year below 25°F a balance point of 24°F is satisfactory. The selection of the solar collectors is based primarily on the efficiency over the average operating temperature range, ease of installation, and aesthetics. As mentioned in Phase II, the KTA tubular collector has the ability to compensate for off-optimum roof tilt be rotating the individual tubes. This feature not only enhances the aesthetics of the collector array but also reduces cost by eliminating the need for a support superstructure.

In order to determine the collector tilt which maximizes the amount of solar energy utilized on an annual basis the amount of solar energy collected and solar energy utilized was calculated for two different tilts. These calculations are based on the KTA collector efficiency curve. Appendix V shows the procedure used to determine the maximum amount of solar energy collected for the months of January and June for collector tilts of 32° and 42°. Collector output for the remainder of the year was determined in a similar manner.

The amount of solar energy utilized for various collector areas and tilts of 32° and 42° was then determined. The first six columns of the calculation sheets in Appendix VI show the procedure used. Figure 86 shows the results of these calculations for 32° and 42° tilts over a collector area range of 50 to 150 square feet.

The amount of solar energy utilized for two tilt angles is essentially the same over a range of 75 to 115 square feet and only slightly different for larger and smaller areas.

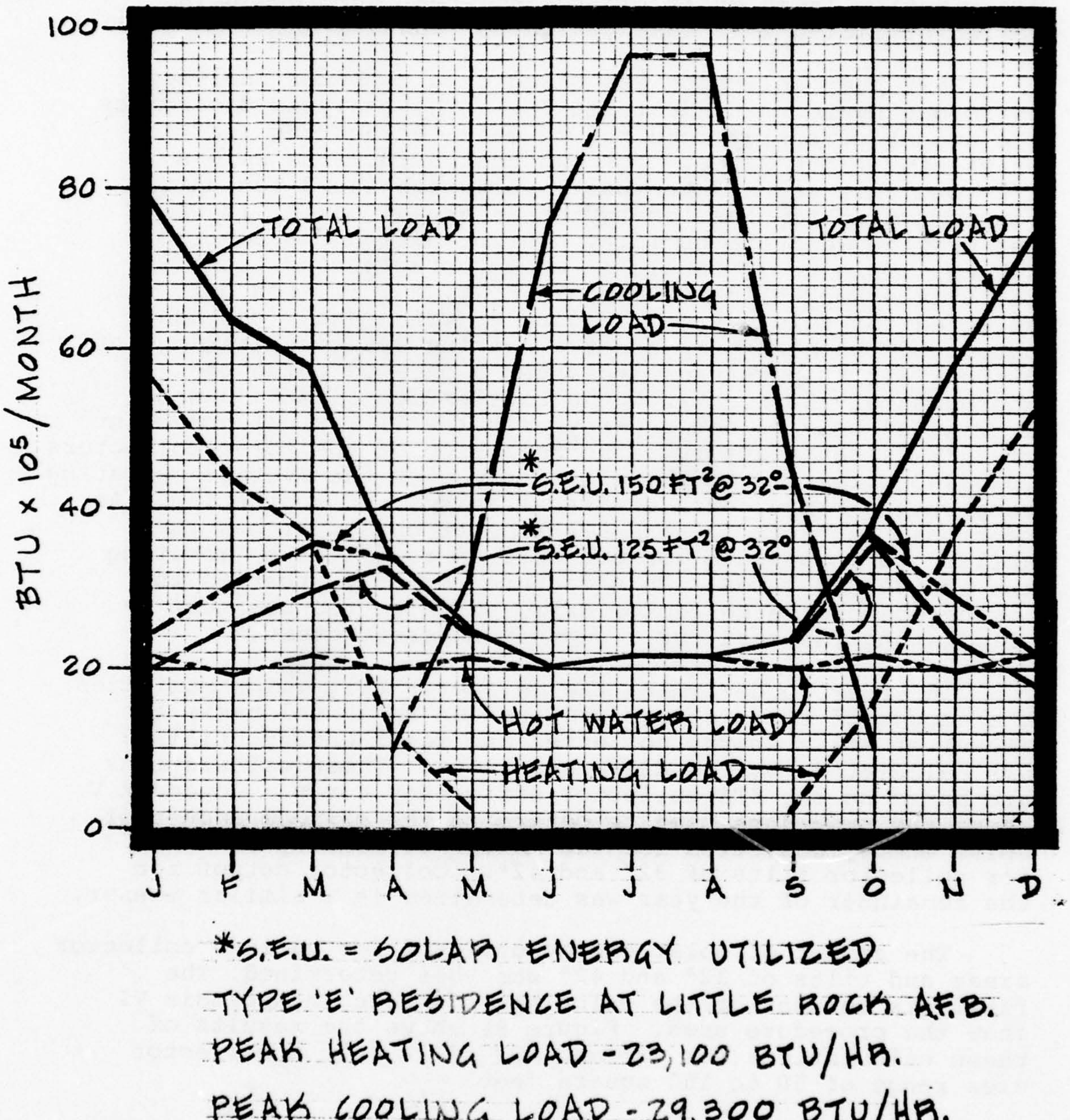


Figure 84. Annual Load Distribution

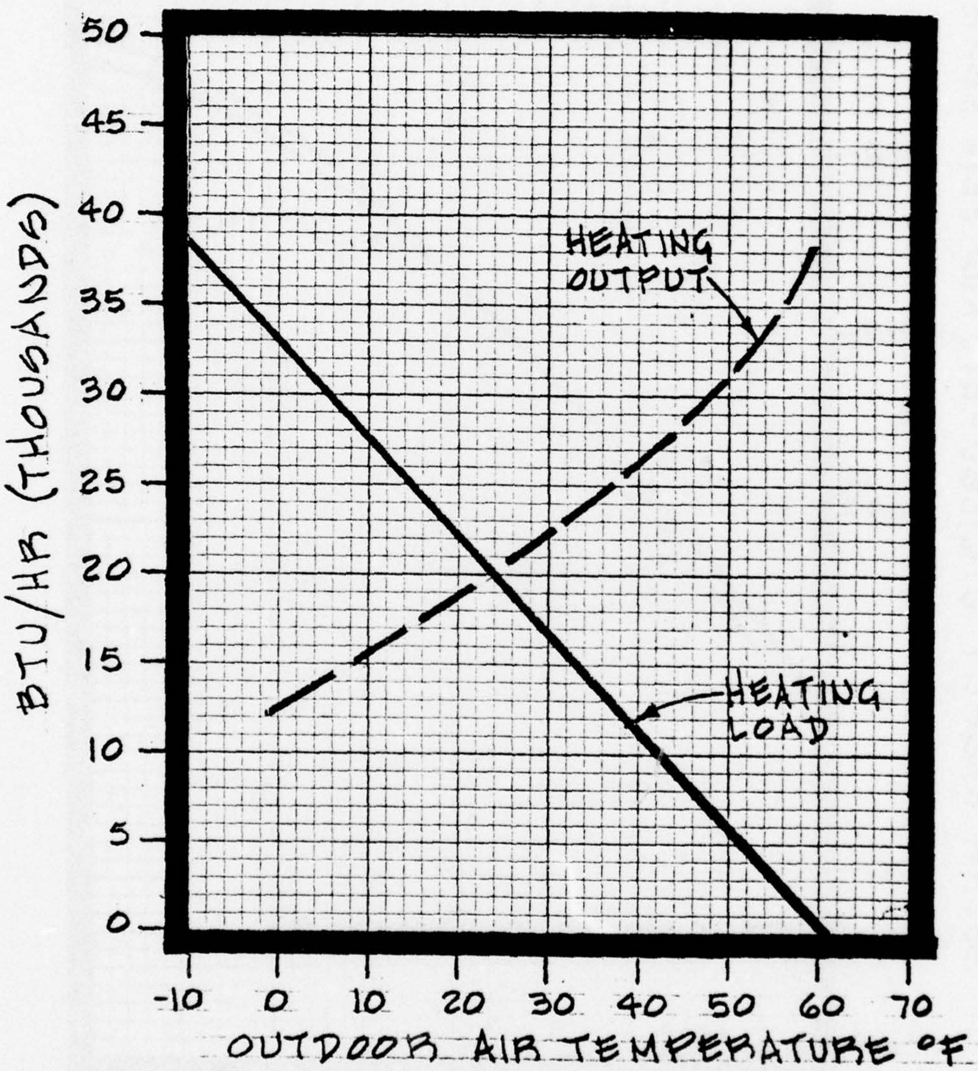


Figure 85. Heat Pump Heating Season Performance

THIS PAGE IS BEST QUALITY PRACTICABLE
FROM COPY FURNISHED TO DDC

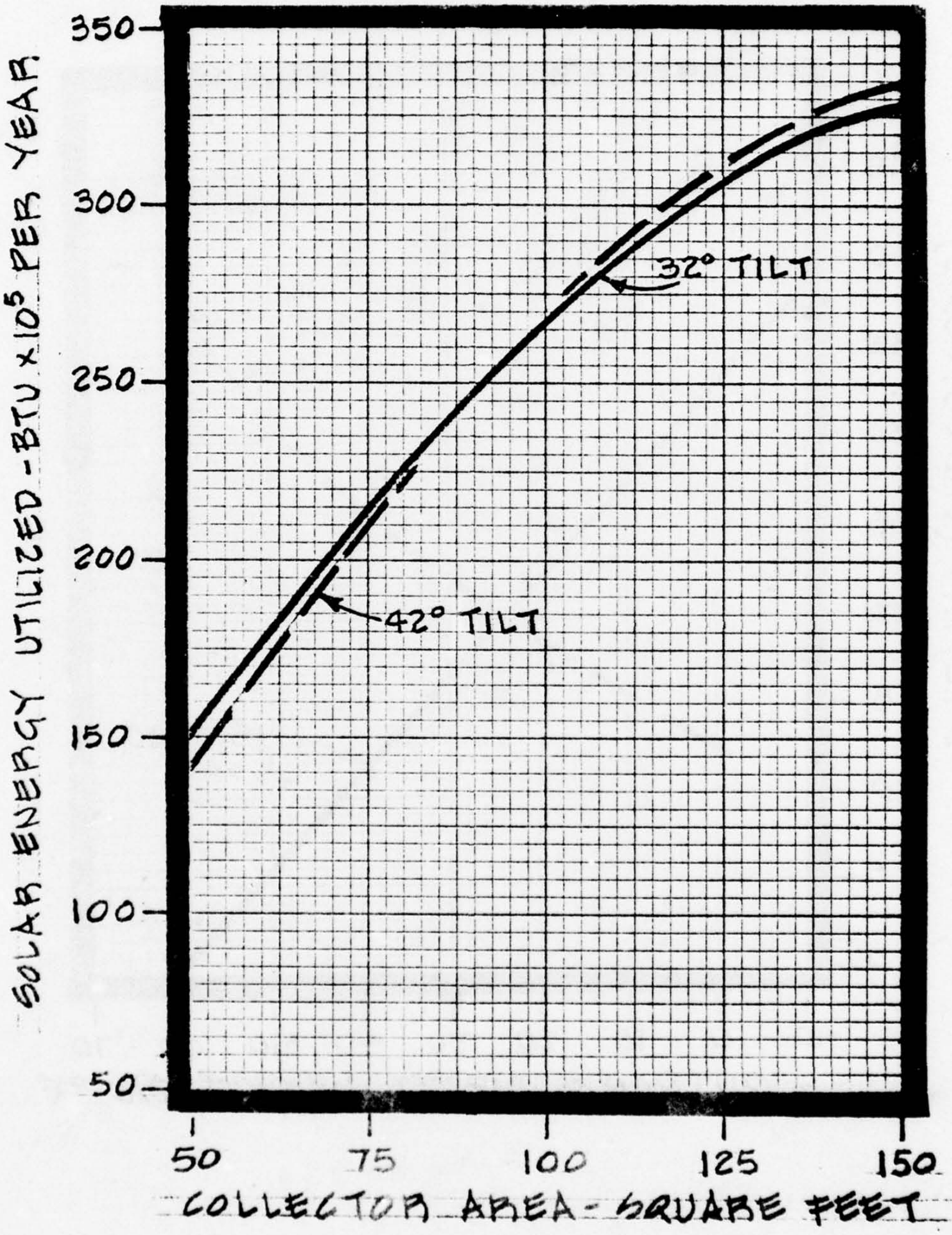


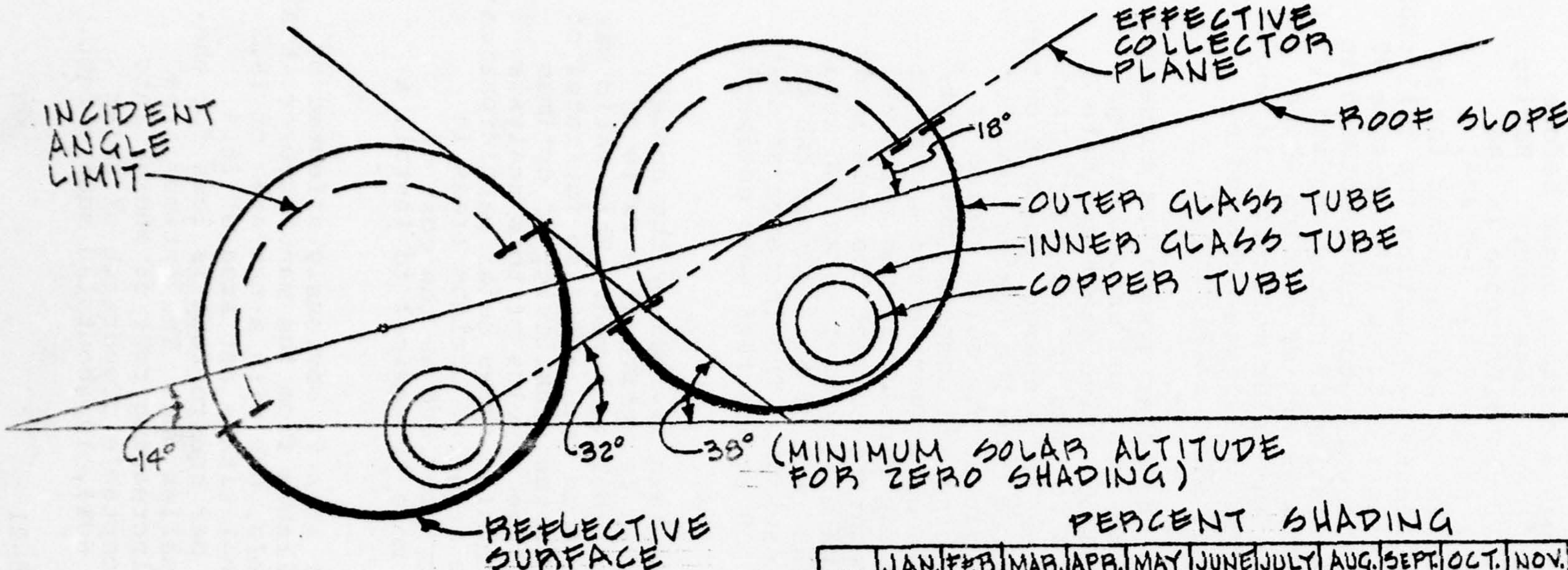
Figure 86. Effect of Collector Tilt on Solar Energy Utilized

The configuration of the KTA collector is such that the tubes can only be rotated through a certain angle before significant shading between adjacent tubes occurs. Having determined that there is very little difference in the amount of solar energy used for tilts of 32° and 42°, a shading analysis of the KTA collector was performed for an effective tilt 32° which is achieved by mounting the collector flat on the 14° roof and rotating the tubes 18°. Figure 87 shows a schematic diagram of the collector tubes rotated in this manner and includes a tabulation of hourly shading percentages for each month. The amount of shading which occurs during the hours of maximum solar collection is not significant. Accordingly, a tilt of 32° was achieved by rotating the collector tubes as shown in Figure 87.

The final step in sizing the solar system was to determine the optimum area based on system cost and annual savings. The calculation procedure used to arrive at an estimate of savings is shown in the last 3 columns of the computation sheets in Appendix VI. The heat pump energy consumption was based on the performance figures from the Lennox computer analysis. Since the heat output of the solar system is being applied to both the heating and domestic hot water loads the savings are determined by summing the energy required to meet these loads for various system sizes and comparing it to the energy required to meet the same loads without the solar system. The energy savings in KWH were then multiplied by the current rate of \$0.0245 per kwh to arrive at the annual dollar savings. The incremental costs of solar systems from 25 to 150 square feet were estimated in 25 square foot increments.

The incremental cost was then divided by the annual savings for each collector area to arrive at a ratio of dollar of incremental cost per dollar saved. This ratio was computed for the current electric rate and also for rates of \$0.5 and \$0.10 per kwh to show the sensitivity of optimum area to the cost of energy. The results of this analysis are shown in Figure 88, which also shows solar participation on the right hand axis. The range of optimum areas is extremely sensitive to energy cost and as the cost of energy increases it becomes more economical to install a larger solar system.

The choice of collector area is obviously affected by the nominal panel sizes available from the manufacturer. KTA has two module sizes available, one with a net area of 18.2 square feet and a larger panel with a net area of 36.4 square feet. The unit cost per square foot is less for the large module than for the smaller one. The optimum area from Figure 80, taking the increasing cost of energy into account and achieving an acceptable proportion of solar participation (greater than 60%), is about 120 square feet.



PERCENT SHADING

	JAN.	FEB.	MAR.	APR.	MAY	JUNE	JULY	AUG.	SEPT.	OCT.	NOV.	DEC.
7	—	—	—	—	—	—	—	—	—	—	—	—
8	47.0	34.2	21.4	9.3	3.8	1.3	3.8	9.3	21.4	34.2	47.0	52.4
9	26.9	13.3	1.3	—	—	—	—	—	1.3	13.3	26.9	32.3
10	11.5	—	—	—	—	—	—	—	—	—	11.5	17.7
11	2.0	—	—	—	—	—	—	—	—	—	2.0	7.5
12	—	—	—	—	—	—	—	—	—	—	—	4.9
1	2.0	—	—	—	—	—	—	—	—	—	2.0	7.5
2	11.5	—	—	—	—	—	—	—	—	—	11.5	17.7
3	26.9	13.3	1.3	—	—	—	—	—	1.3	13.3	26.9	32.3
4	47.0	34.2	21.4	9.3	3.8	1.3	3.8	9.3	21.4	34.2	47.0	52.4
5	—	—	—	—	—	—	—	—	—	—	—	—

Figure 87. Effect of Collector Tube Rotation on Shading

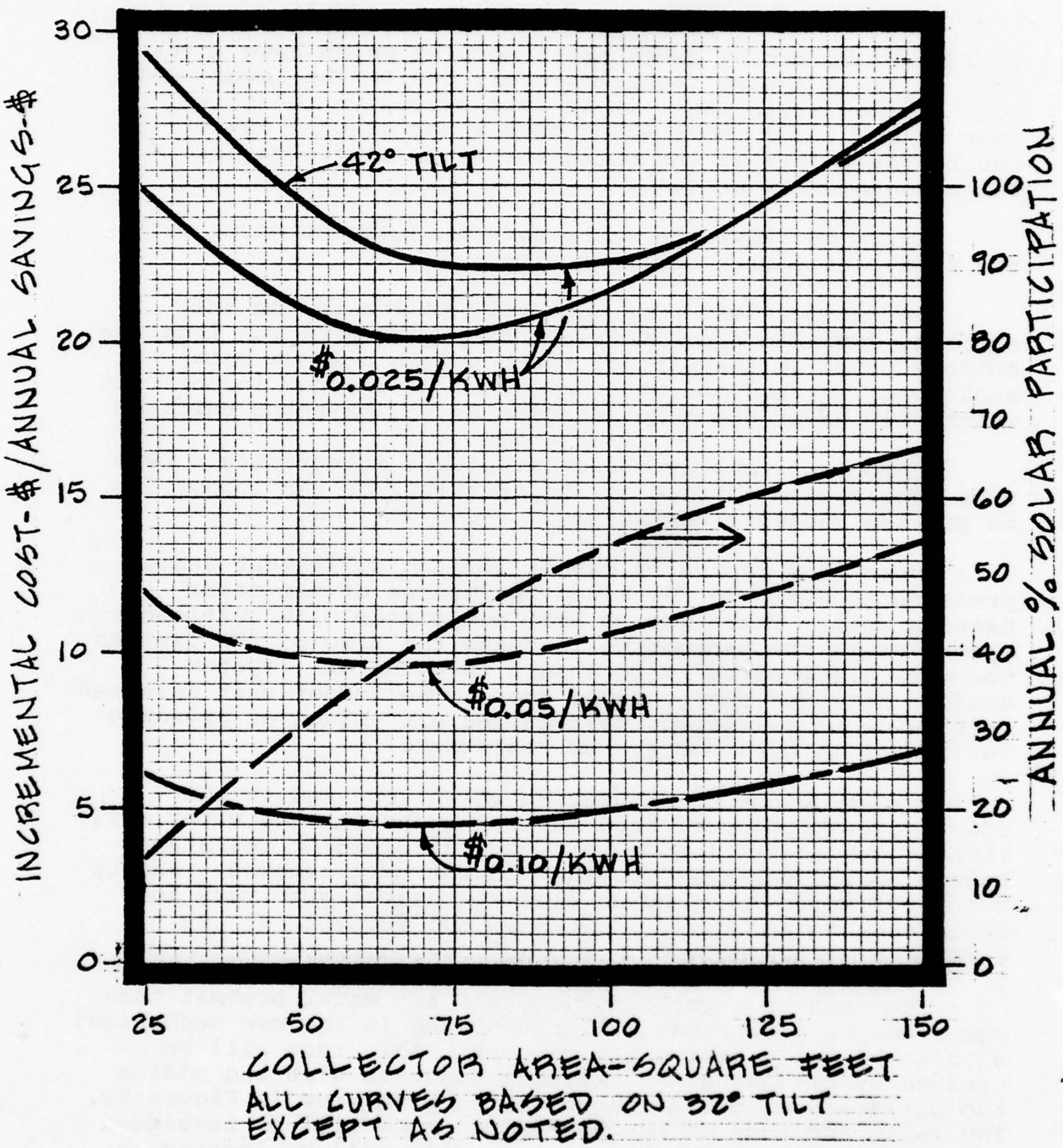


Figure 88. Cost Effectiveness VS. Collector Area

This could be met by seven small modules which have a net area of 127.4 square feet. However, due to the lower unit cost of the larger panel it is less expensive to install four of the large units which have a net area of 145.6 square feet. This is the collector array configuration selected for this project.

Figure 89 shows the south elevation of the house with the four panel array configuration.

The outdoor heat pump unit will be located on the existing concrete pad. Refrigerant piping will run from the outdoor unit, up the side of the house in a sheet metal enclosure, through the attic and down to the new indoor unit which will be in the same location as the existing unit.

A problem commonly encountered when integrating a heat pump with a solar system is the inability of the indoor fan to provide adequate air volume.

These fans typically cannot develop sufficient static pressure to overcome the added resistance of the solar heating coil. There are no heat pumps made with sufficient space inside the cabinet to increase the size of the fan; so the only alternative is to ensure that the air system design minimizes the pressure drop. The indoor unit selected is capable of providing the required air volume at pressure sufficient to overcome system resistance.

The component configuration which resulted in the lowest static pressure drop is with the solar hot water coil located inside the indoor unit and the electric resistance backup heater normally provided in the unit removed. Backup heat will be provided by an electric heater with very low resistance to air flow, located in the ductwork at the discharge of the unit.

The main storage tank, domestic hot water preheat tank, pumps and control panel will be located in the new mechanical room to be built behind the carport. This room will be created by extending the existing concrete slab and adding two walls and a roof as shown in the plan view in Figure 89. The walls and roof of the mechanical room will be insulated and a small heater set to maintain 40°F will be located in the room to prevent freezing in the event the solar system is inoperative.

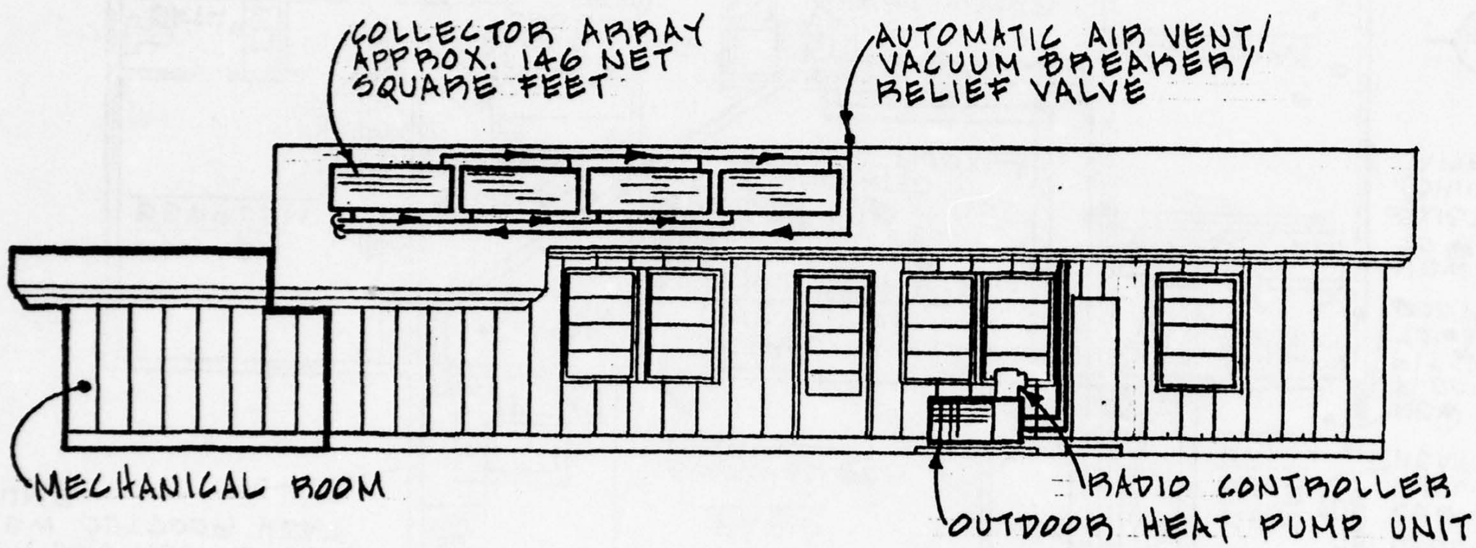


Figure 89. South Elevation #170 Alabama

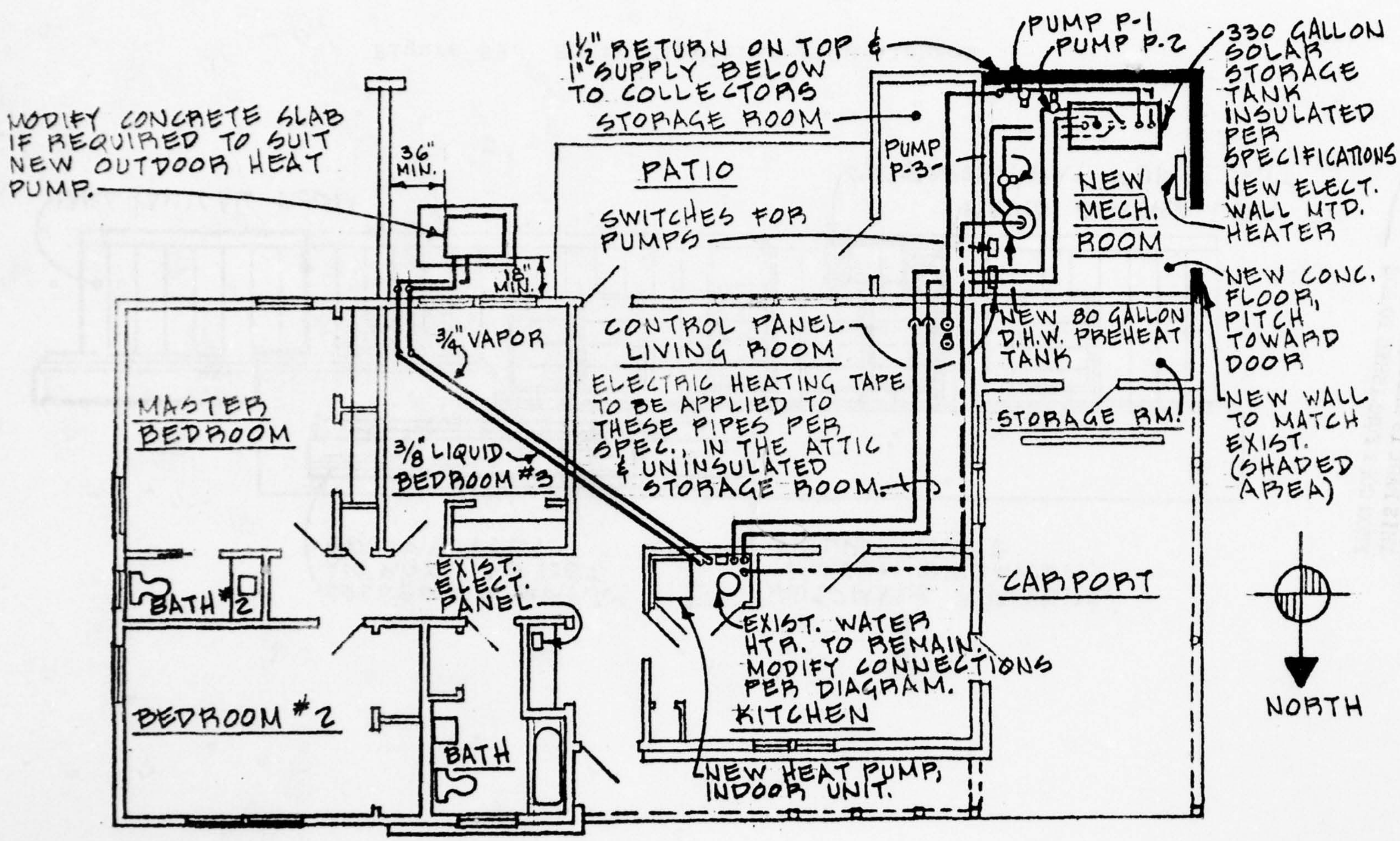


Figure 90. Plan View - Equipment Location

The main storage tank is a conventional above ground oil storage tank with a total capacity of 330 gallons. Sufficient space will be left in the tank to drain the collectors and array piping.

The domestic hot water piping will be rerouted as shown in Figure 90. The cold water make up will be connected to the new 80 gallon preheat tank located in the mechanical room. Hot water will flow from the preheat tank to the existing hot water heater. A tempering valve will be provided at the discharge of the existing tank to prevent scalding at taps.

System control devices will be provided in a single control panel with all temperature and temperature differential settings adjustable on the face of the panel. The controls manufacturer will provide installation instructions and a wiring diagram.

The need for water treatment should be investigated as outlined in the specification. The system uses all copper piping and the storage tank is lined with a protective coating. In addition, after initial filling, no new fresh water is introduced into the system. This will minimize the possibility of scale formation. However, it is recommended that water quality be analyzed by a qualified lab to determine if any pH control or other additives are necessary.

To determine the effectiveness of the solar system/heat pump combination it is recommended that individual electric meters be installed to measure the electrical energy consumed by the heat pump indoor and outdoor units, the electric water heater, and the three systems pumps. Arkansas Light and Power may be willing to participate in the instrumentation portion of the installation by providing the meters.

6. EQUIPMENT LIST

The following list gives a brief description of the major pieces of equipment.

Solar Collector:

- Tubular, semi-concentrating
- Tubes rotated 18° to give effective tilt angle of 32°
- Net aperture approximately 150 SF-Minimum 140 SF

The main storage tank is a conventional above ground oil storage tank with a total capacity of 330 gallons. Sufficient space will be left in the tank to drain the collectors and array piping.

The domestic hot water piping will be rerouted as shown in Figure 90. The cold water make up will be connected to the new 80 gallon preheat tank located in the mechanical room. Hot water will flow from the preheat tank to the existing hot water heater. A tempering valve will be provided at the discharge of the existing tank to prevent scalding at taps.

System control devices will be provided in a single control panel with all temperature and temperature differential settings adjustable on the face of the panel. The controls manufacturer will provide installation instructions and a wiring diagram.

The need for water treatment should be investigated as outlined in the specification. The system uses all copper piping and the storage tank is lined with a protective coating. In addition, after initial filling, no new fresh water is introduced into the system. This will minimize the possibility of scale formation. However, it is recommended that water quality be analyzed by a qualified lab to determine if any pH control or other additives are necessary.

To determine the effectiveness of the solar system/heat pump combination it is recommended that individual electric meters be installed to measure the electrical energy consumed by the heat pump indoor and outdoor units, the electric water heater, and the three systems pumps. Arkansas Light and Power may be willing to participate in the instrumentation portion of the installation by providing the meters.

6. EQUIPMENT LIST

The following list gives a brief description of the major pieces of equipment.

Solar Collector:

- Tubular, semi-concentrating
- Tubes rotated 18° to give effective tilt angle of 32°
- Net aperture approximately 150 SF-Minimum 140 SF

- Minimum instantaneous performance curve: efficiency vs (temperature differential (collector-ambient) divided by insolation - described by the linear equation y = 0.67 -0.34x over an abscissa range of 0 to 0.6.
  (y intercept = 0.67, slope -0.34)
- Manufacturer - KTA Corporation Model - 85 or approved equal.

Heat Pump

- Split system with solar heating coil mounted inside indoor unit - electric resistance coil removed.
- ARI Standard 240 ratings.
  cooling capacity - 30,500 BTU/hr.
  high temperature heating capacity - 31,000 BTU/hr
  low temperature heating capacity - 18,000 BTU/hr
- Solar coil - minimum capacity 23,000 BTU/hr
  7.5 gpm water @115°F
  1200 cfm air @ 70°
  maximum static pressure drop - 0.25 inches w.g.
- Manufacturer - Lennox HP10-311 outdoor unit, CBP 10-41 indoor unit or approved equal

Electric Heating Coil;

- U.L. labelled, slip-in, open coil
- One step of 7KW with elements derated to maximum of 25 watts per square inch of element surface
- Maximum static pressure drop - 0.06 inches w.g.
- Manufacturer - INDEECO or approved equal

Storage Tank:

- Carbon steel "Obround" approximate dimensions 27" wide x 44" high x 72" long - capacity approx. 330 gal.
- Manhole and fittings as shown on drawing
- Pressure tested to 5 psi
- Interior painted with one coat of Apexior Number 1 exterior painted with one coat red oxide primer
- Insulated with 2 inches semi-rigid fiberglass covered with vinyl coated facing

Domestic Hot Water Preheat Tank:

- 80 gallon vertical tank - stone lined - fiberglass insulation sheathed with sheetmetal with enamel finish
- Integral finned copper water coil heat exchanger with minimum 15 S.F. surface area, 3.5 gpm flow rate.
- Manufacturer - Ford Products Corporation Model TC80 or approved equal.

Circulation Pumps:

- In line, centrifugal with 1/12 HP integral electric motor.
- Variable head adjustment
- Manufacturer - Grundfos Pumps Corp. Model UP26-64 or approved equal
- P-1 - Collector pump - 4 gpm @ 14 Feet head
- P-2 - Heating pump - 7.5 gpm @ 15 feet head
- P-3 - DHW pump - 3.5 gpm @ 10 feet head

Controls;

- Two differential controllers
- Outdoor reset control
- Adjustable set points and differentials
- All components except sensors mounted in prewired, enclosed panel
- Manufacturer - Natural Power Inc. Model S10-101 or approved equal.

5.7 COST ESTIMATE

System costs are based on the equipment list in the previous section. Prices have been obtained from the manufacturers listed where specific equipment models are indicated. Cost for piping/valves/installation etc. were obtained from the 1977 Means Construction Costs. Table 15 shows the estimated installed costs.

5.8 ECONOMIC ANALYSIS

The $1800 heat pump cost was deducted from the total system cost of $10,000 for economic analysis of the solar system alone. The analysis is based on a net increase in energy cost of 5% per year after inflation and a present energy cost of $0.025 per KWH. The cost of maintenance on the solar system was assumed to be negligible. The cost of the heat pump maintenance was assumed the same for a system with solar as it is for a system without solar. Therefore, the cost of heat pump maintenance would not enter into the economic analysis.

Table 16 shows that the cumulative cash flow for the system that satisfies the heating load first and the domestic hot water second. The system has a 32 year pay back.

Table 17 shows that the cumulative cash flow for the system that satisfies the domestic hot water load first and the heating load second. The system has a 27 year pay back.

Based on the above economic analysis the system installed at Little Rock AFB should satisfy the domestic hot water requirement first and the heating load second.

Table 15. ESTIMATED INSTALLED COST

House Modifications - Mechanical Room	$1,100
Collectors	1,950
Storage Tank	390
Pumps	695
Heat Pump	1,800
Controls	800
Electric Heating Coil	500
DHW Preheat Tank	300
Piping	1,995
Pipe Insulation	570
Total	$10,100
Minus Heat Pump Cost	-1,800
Total Cost For Solar System	$8,300

Table 16. CUMULATIVE CASH FLOW

(HEATING LOAD SATISFIED FIRST)

YEAR	INVESTMENT	ANNUAL SAVINGS	CUMULATIVE CASH FLOW
1	$8,300	$115	-8,185
2	0	121	-8,064
3	0	127	-7,937
4	0	133	-7,804
5	0	140	-7,665
6	0	147	-7,518
7	0	154	-7,364
8	0	162	-7,202
9	0	170	-7,032
10	0	178	-6,854
15	0	228	-5,818
20	0	291	-4,497
25	0	371	-2,811
30	0	473	- 660
31	0	497	- 163
32	0	522	+ 359

Table 17. CUMULATIVE CASH FLOW

(DOMESTIC HOT WATER SATISFIED FIRST)

YEAR	INVESTMENT	ANNUAL SAVINGS	CUMULATIVE CASH FLOW
1	$8,300	$150	-8,150
2	0	158	-7,992
3	0	166	-7,826
4	0	175	-7,651
5	0	184	-7,467
6	0	194	-7,273
7	0	204	-7,069
8	0	215	-6,854
9	0	226	-6,628
10	0	238	-6,390
15	0	306	-5,004
20	0	394	-3,218
25	0	508	- 917
26	0	534	- 884
27	0	562	+ 179

APPENDIX I REFERENCES

SOLAR/HEAT PUMPS

AMBROSE, E.E., "The Heat Pump: Performance Factor, Possible Improvements" Heating/Piping/Air Conditioning, May, 1974.

AMBROSE, E.R., Heat Pumps and Electric Heating, John Wiley & Sons, Inc., 1966.

AMBROSE, E.R., "Heat Pump System Employing Storage and using Auxiliary Heat".

AUSTERWIEL, Louis, "Optimized Data for Heat Pump Systems", Building Systems Design, June/July, 1974.

AVERY, Lester T., "Use Hot Gas in Hot Deck of Dual Duct Heat Pump System", Heating, Piping & Air Conditioning, April, 1966.

BALASUBRAMANIAM, M., and ACHRENK, G.L., "Heat Pumps, Thermal Conditioning and Energy Conservation", National Center for Energy Management and Power, University of Pennsylvania.

BURGER, R.J., "Heat Pump provides Year-Round Conditioning for Kyoto Office Building", Electric Ligh and Power, March, 1948.

CALVERT, F.O., and HARDEN, D.G., Hybrid Solar Energy and Electric Heat Pump, ASME publication 73WA/Sol-5.

CHAN, D.C., "Residential Energy Consumption and Small-Scale Options of Energy Systems for Space Heating", The Mitre Corporation, Bedford, Massachusetts, MTR-2951, November, 1974.

COOGAN, Charles H., Jr., The Residential Heat Pump in New England, Connecticut Light and Power Company, August, 1948.

COTTINGHAM, J.G., and WILHELM, W. G., "Nighttime Use of Flat Plate Type Solar Energy Collectors as a Heat Sink in 'Solar Assisted Heat Pump' Systems. A Preliminary Analysis", Brookhaven National Laboratory for the ERDA, September 9, 1975.

DOUGLASS, Edwin S., "Innovations in Unitary Heat Pump Designs", ASHRAE Preprint, Symposium #2, page #3.

DUNNING, R.L. and Geary, Dr. L.C., "Analysis of the Efficiency and Cost of Equivalent Residential Comfort - Conditioning Systems", The Energy Utilization Project, Power Systems Planning, Westinghouse Electric Corporation, East Pittsburgh, Pa., Report #PSP 2-10-74.

DUNNING, R.L., DEARY, Dr. L.C. and TRUMBOWER, S.A., "Analysis of Relative Efficiencies of Various Types of Heating Systems". The Energy Utilization Project, Power Systems Planning, Westinghouse Electric Corporation, East Pittsburgh, Pa., Report #PSP 10-3-73 (Revised 1-21-74).

EBAUGH, Newton C., "Engineering Performance of a Residential Air-to-Air Heat Pump", Building Systems Design, May, 1971.

FARR, Arthur H., "The Heat Pump Primer - Part 1, Part 2, Part 3, Part 6", The Refrigeration & Air-Conditioning Business, March, 1959, May, 1959, July, 1959, January, 1960.

FISCHER, H.C., "The Annual Cycle Energy System", report of research by Oak Ridge National Laboratory, Oak Ridge, Tennessee, sponsored by U.S. Dept. of HUD and the FEA under Union Carbide Corporation Contract with ERDA.

GILMAN, Stanley F. (Coordinator), Solar Energy Heat Pump Systems for Heating and Cooling Buildings, ERDA Doc. COO-2560-1, Con. 7506130, 1975.

"Heat-Pump Reliability Shows Big Gains", Electrical World, August 1, 1973.

Heat Pumps - Application & Reliability Symposium papers for ASHRAE at Nassau, Bahamas, June 25-29, 1972.

Heat Pumps - Improved Design and Performance, Symposium papers for ASHRAE at San Francisco, California, January 19-22, 1970.

"Improving Heat Pump Acceptability", ASHRAE preprint, Symposium #2, paper #4, Dallas, January, 1976.

JARDINE, Douglas M., "A Report on Integrating Solar Energy with Heat Pumps to Conserve Fossil Fuels and Maintain Satisfactory Electric Power System Capacity Factors", K-76-55U(R), Kaman Sciences Corporation, Colorado Springs, Colorado, 26 April, 1976.

JARDINE, Douglas M., "Report of the Phoenix Corporation-City of Colorado Springs Solar Heating Project", prepared by Kaman Sciences Corporation for the ERDA.

JASPERS, H.A. and du PRE', F.K., "Stirling Engine Design Studies of An Underwater Power System and a Total Energy System", Philips Laboratories, A division of North American Philips Corporation, Briarcliff Manor, N.Y., 739035.

JORDAN, Richard C. and LIU, Benjamin Y.H.. "The ASHRAE Monograph on Applications of Solar Energy for Heating and Cooling Buildings", ASHRAE preprint, Symposium #12, paper #2.

MACRISS, Robert A., "Selecting Refrigerant-Absorbent Fluid Systems for Solar Energy Utilization", Institute of Gas Technology, Chicago, Illinois, paper presented at ASHRAE Semi-annual Meeting, Dallas, Texas, February 1-5, 1976.

MILNARK, Sam, "Evaluate the Air-Cycle Machine as a Heat Pump", Air Conditioning & Refrigeration Business, November, 1975.

MILLER, Jay F., "The Decentralized Heat Pump Approach", ASHRAE preprint, Symposium #18, paper #4.

NELSON, Lorne W. and TOBIAS, James R., "Energy Savings in Residential Buildings, ASHRAE Journal, February, 1974.

OLIVERSON, Robert L., "Engineering Basics of Heat Pumps and Their Uses", Specifying Engineer, July, 1976.

PETERSEN, Stephen R., "Economic Optimization in the Energy Conservation Design of Single-Family Housing", ASGRAE preprint, Symposium #DA-76-4, paper #3, October 15, 1975.

RATAI, W.R., "Heat Pump/Recovery System Conserves Energy, Cuts Costs", Actual Specifying Engineer, April, 1972.

RATAI, Walter R., "Saving Energy with Heat Pumps, Recovery Storage", Specifying Engineer, July, 1976.

REISTAD, G. M., "Availability Analysis of the Heating Process and a Heat-Pump System".

SEGERSTROM, Stewart, "Heat Pumps Today", ASHRAE Journal, July, 1971.

"Singer Markets Heat Pump Valves", Air Conditioning & Refrigeration Business, March, 1976, page 13.

SMITH William T., "Energy Use in Military Housing", Air Conditioning & Refrigeration News, August 13, 1973.

Solar Applications of ES Absorption Chillers, York Division Borg-Warner, Form 155.16-TD.

TRELEASE, Stephen W., "Electro-Hydronic Heat Recovery", Building Systems Design, May, 1971.

UCAR, Manas and DRUCKER, Eugene E., "Computer Model for a Solar Assisted Closed Liquid Loop Heat Pump System, ASHRAE Preprint, Symposium #9, paper #4.

WHITEHEAD, E.R. and ROLEY, R.D., "The Heat Pump - A Proven Device for Heat Recovery Systems", ASHRAE preprint, Symposium DA-76-2, paper #2.

WOLF, Martin, "The University of Pennsylvania Solar Heating/Cooling System Program:, ASHRAE preprint, Symposium #9, paper #3.

WURM, Jaroslav and PANIKKER, Gopal, P.K., "Evaluation of Engine-Driven Heat Pump Systems of Small Capacities", Institute of Gas Technology, Chicago, Illinois, paper presented at Environment Energy Conservation Symposium of the EPA and the ERDA, Denver, Colorado, November 3-6, 1975.

YANAGIMACHI, Masonosuki, "Experimental Data on Annual Electric Consumption and Cost for all Electric 'Yanagimachi Heat Pump House III'", May, 1973.

YANAGIMACHI, Masonosuki, "Air Source Heat Pump-Heat Storage System of Hiroshima Regional Station of the Japan Broadcasting Corporation for Year-Round Air-Conditioning", Transactions - Shase - Japan, Vol. 3., 1965.

YANAGIMACHI, Masonosuki, "Report on Two and a Half Years Experimental Living in Yanagimachi Solar House II", United Nations Conference on New Sources of Energy E/CONF. 35/S/94, 23 May, 1961.

YELLOTT, John I., "Japanese Sun House: Milestone in use of Solar Energy", Western Architect and Engineer, McGraw-Hill Co. of California, March, 1960.

York Heat Pump Application Manual, A.I.A. File No. 30-F-21.

STORAGE

BELTON, Geoffrey and AJAMI, Fouad, "Thermochemistry of Salt Hydrates", University of Pennsylvania, National Center for Energy Management and Power and Towne School of Civil and Mechanical Engineering, Philadelphia, Pa., Report No. NSF/ RANN/SE/GI27976/TR/73/4, May 1973.

DUDLEY, James C., "Thermal Energy Storage Unit for Air Conditioning Systems Using Phase Change Material", University of pennsylvania, National Center for Energy Management and Power and Towne School of Civil and Mechanical Engineering, Philadelphia, Pa., Report No. NSF/RANN/SE/GI27976/TR72/8, August, 1972.

KAUFFMAN, Kenneth and Pan, Yen-Chi, "Thermal Energy Storage in Sodium Sulfate Decahydrate Mixtures", University of Pennsylvania, National Center for Energy Management and Power and Towne School of Civil and Mechanical Engineering, Philadelphia, Pa. Report No. NSF/RANN/SE/GI/27976/TR72/11, December, 1972.

KAUFFMAN, Kenneth and GRUNTFEST, IRVING, "Congruently Melting Materials for Thermal Energy Storage", National Center for Energy Management and Power, University of Pennsylvania, Philadelphia, Pa., Report No. NCEMP-20, November 1973.

KAUFFMAN, Kenneth and PAN, Yen-Chi, "Congruently Melting Materials for Thermal Energy Storage in Air Conditioning", University of Pennsylvania, National Center for Energy Management and Power and Towne School of Civil and Mechanical Engineering, Philadelphia, Pa. Report No. NSF/RANN/SE/GI27976/TR73/5, June, 1973.

LORSCH, Harold G., "Latent Heat and Sensible Heat Storage for Solar Heating Systems", University of Pennsylvania, National Center for Energy Management and Power, Philadelphia, Pa., Report No. NSF/RANN/SE/GI27976/TR72/20, December, 1972 (Revised May, 1974).

"Presentation; HUD/MIUS (Modular-Sized Integrated Utility Systems) Technology Briefing, NASA - Manned Spacecraft Center, Houston, Texas, November 2, 1972", Artech Corp., Falls Church, Virginia.

SHELTON, Jay, "Underground Storage of Heat in Solar Heating Systems".

SEGASER, C.L., "Technological Evaluation of Thermal Energy Storage for MIUS Program, Oak Ridge National Laboratory, Union Carbide Corporation, Nuclear Division, Oak Ridge, Tennessee, February, 1973.

APPENDIX II

LETTER OF INQUIRY TO MANUFACTURERS

Re: Solar Assisted Hybrid Heat Pumps
for the USAF

Gentlemen:

As discussed during our telephone conversations, we have been retained by the USAF to prepare a feasibility study for the application of solar assisted hybrid heat pump systems for heating and cooling houses.

Initially the study would be limited to existing housing at Little Rock AFB, Arkansas, and Scott AFB, Illinois, but would show the sensitivity to other climates.

This phase of the study calls for us to identify at least three manufacturers who can presently assemble a hybrid heat pump from their standard components or who can readily develop this capability.

For the purpose of this project, the definition of a hybrid heat pump is a machine which combines water/air and air/air characteristics and hardware and which can utilize both water and outdoor air as a heat source, although not simultaneously. The heat pump must also be capable of operating in a cooling mode using the outdoor coil as a condenser.

For your information the heat pump will work in conjunction with a liquid solar energy system using water as the thermal storage medium. We envisage that the heating modes of operation would be as follows:

a. Direct use of solar heated storage water through a coil in the air supply duct (water above 100°F).

b. Indirect use of solar heated water storage as a heat source for the heat pump in water/air configuration (water between 100°F and 60°F).

c. Heat pump in air/air configuration using outdoor air as a heat source.

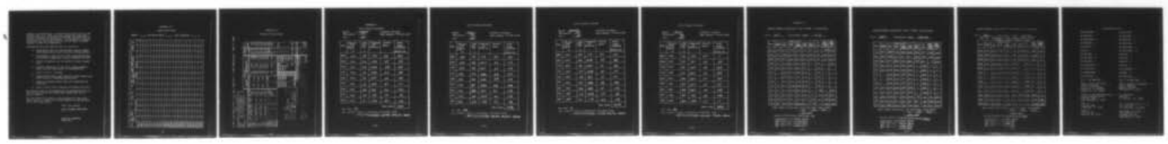
d. Direct electric resistance heating when solar storage is depleted and the outdoor air temperature is too low for the heat pump to meet the load.

Mode (a) will always be used in preference to any other mode. It is probable that the most effective yearly performance will be obtained by using mode (c) whenever the

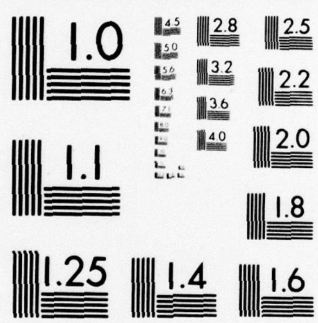
AD-A058 626 DUBIN-BLOOME ASSOCIATES NEW YORK F/G 13/1
SOLAR ASSISTED HEAT PUMP STUDY FOR HEATING OF MILITARY FACILITI--ETC(U)
JUL 78 F L BEASON, L W STROTHER F08635-76-C-0276

UNCLASSIFIED AFCEC-TR-78-6 NL

3 OF 3
AD
A058626



END
DATE
FILMED
11-78
DDC



MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

outdoor air temperature is sufficiently high to give a COP of more than 2.5. Mode (b) using stored low grade heat would be reserved for periods of extreme cold when the COP of mode (c) would be unfavorable. Our analysis of yearly loads made for the study would of course serve to either confirm or deny this hypothesis.

Information we would require from you would be:

1. Performance data for varying heat source temperature water-to-air and air-to-air configurations.

2. Diagrammatic layout of the heat pump identifying the standard components used to assemble the unit.

3. Method of defrost and its effect on total energy use.

4. Overall dimensions of the unit, access requirements and any constraints on installatoins.

5. Cost of unit to customer.

6. Predicted delivery lead time for single orders and batch orders of ascending quantities.

7. Range of standard sizes that will eventually be available.

8. Level of R&D effort required to produce the unit.

We would be glad of the opportunity of working with you on this project and are sure that the results will be of mutual benefit.

Our time frame for analysis and production of the first phase report is tight and we look forward to hearing from you in the near future.

Very truly yours,

DUBIN-BLOOME ASSOCIATES,
P.C.

Barry D. Symmonds
Associate

APPENDIX III

Sample Data Sheet

MONTH: ______ COLLECTOR AREA: ________ SKY CONDITION: __________

TIME	AVG COLL TEMP °F	AMB TEMP °F	I BTU/SF	Q_{COLL} BTU/HR	HTG LOAD	NET Q	ΔT_C	MODE OF OPER	HP POTENT OUTPUT BTU/HR	LOAD / OUTPUT	POWER INPUT W/HR	POWER CONSUM.	COP	COP-1 / COP	HTG Q
0800															
0900															
1000															
1100															
1200															
1300															
1400															
1500															
1600															
1700															
1800															
1900															
2000															
2100															
2200															
2300															
2400															
0100															
0200															
0300															
0400															
0500															
0600															
0700															

APPENDIX IV

Heating Cooling Load

USAF HEAT PUMP STUDY LITTLE ROCK ARKANSAS — LOAD CALCULATIONS - TYPE 'E' HOUSE

DESIGN COND	SUMMER DB	SUMMER WB	SUMMER %RH	SUMMER GR/#	WINTER DB	WINTER GR/LB
OUTSIDE	94	79	52	127	20	
ROOM	75	63	50	65.5	70	
DIFFERENCE	19			61.5	50	

						COOLING					HEATING		
						SOLAR		TRANSMISSION					
ITEM	EXP	DIMEN.	AREA	DEDUCT	NET AREA	FACTOR	BTU/HR	TD	"U" FACTOR	BTU/HR	TD	U FACTOR	BTU/HR
WALL	N	46'X8'2"	375	43	332			30	0.055	548	50	.055	913
GLASS					43	33	1 419	19	0.9	775	50	.9	194
WALL	S	46'X8'2"	375	65	310			48	0.055	818	50	.055	852
GLASS					65	84	5 460	19	0.9	1112	50	.9	2 925
WALL	E	28'X8'2"	229	28	201			30	0.055	332	50	.055	553
GLASS					28	33	924	19	0.9	479	50	.9	1260
WALL	W	28'X8'2"	229	19	210			21	0.055	820	50	.055	578
GLASS					19	152	2 888	19	0.9	325	50	.9	855
FLOOR					148 LF						31	BTU/LF	4588
CEILING					1202			39	0.033	1 547	50		1983
ROOF													
PARTITION													
						TOTAL	10 691			6 716			14701
						LATENT HEAT			SOLAR	10 691			
						FACTOR	BTU/HR		FACTOR	BTU/HR			
PEOPLE	4					200	800		250	1000			
LIGHTS	644 WATTS								3.4	2190			
SMALL APPLIANCES										1200			
INFILTRATION	118 CFM X 19 T.D. X								1.08	2421	50	1.08	6 372
INFILTRATION	118 CFM X 61.5 GR/LB X					0.68	4935						
			ROOM LATENT HEAT				5735		Rm Sen. Ht	24 218	SUB-TOTAL		21073
										5735			
SUBTOTAL													
10% DUCT													2107
TOTAL										29 953			23180

APPENDIX V

Solar Energy Collected

MONTH: JANUARY

% SUNSHINE: 44

TILT: 32^{0}

COLLECTOR EFFICIENCY CURVE EQUATION: $Y = 0.67 - 0.34X$

TIME	AVG. COLL. TEMP. $\left(\frac{T_I + T_O}{2}\right)$ °F	AMB. TEMP (T_A) °F	INSULATION (I) BTU/SF.HR	INCIDENT ANGLE	SOLAR ENERGY COLLECTED BTU/SF·HR
0800	120	33	106	62	22
0900	120	36	193	48	74
1000	120	40	256	36	128
1100	120	44	295	25	168
1200	120	48	308	20	182
1300	120	52	295	25	171
1400	120	56	256	36	132
1500	120	55	193	48	78
1600	120	51	106	62	25
				DAILY TOTAL	980

NO. DAYS: 31

AVG. OUTPUT = # DAYS × (% SUNSHINE + 0.1) × DAILY TOTAL

= 31 (.44 + 0.1) 980 = 16405 BTU/SF MONTH

Solar Energy Collected

MONTH: JUNE

% SUNSHINE: 72

TILT: 32^0

COLLECTOR EFFICIENCY CURVE EQUATION: $Y = 0.67 - 0.34X$

TIME	AVG. COLL. TEMP. $\left(\frac{T_1 + T_0}{2}\right)$ °F	AMB. TEMP (T_A) °F	INSULATION (I) BTU/SF.HR	INCIDENT ANGLE	SOLAR ENERGY COLLECTED BTU/SF·HR
0800	145	70	143	63	69
0900	145	73	204	49	112
1000	145	76	251	37	145
1100	145	79	282	28	167
1200	145	82	292	24	174
1300	145	84	282	28	168
1400	145	88	251	37	149
1500	145	90	204	49	118
1600	145	89	143	63	76
				DAILY TOTAL	1178

NO. DAYS: 30

AVG. OUTPUT = # DAYS X (% SUNSHINE +0.1) X DAILY TOTAL

$= 30(.72 + 0.1)1178 = 28{,}979$ BTU/SF MONTH

Solar Energy Collected

MONTH: JANUARY
% SUNSHINE: 44
TILT: 42^0

COLLECTOR EFFICIENCY
CURVE EQUATION: $Y = 0.67 - 0.34X$

TIME	AVG. COLL. TEMP. $\left(\frac{T_1 + T_0}{2}\right)$ °F	AMB. TEMP (T_A) °F	INSULATION (I) BTU/SF.HR	INCIDENT ANGLE	SOLAR ENERGY COLLECTED BTU/SF·HR
0800	120	33	116	59	25
0900	120	36	206	44	80
1000	120	40	269	31	135
1100	120	44	308	18	177
1200	120	48	321	10	191
1300	120	52	308	18	180
1400	120	56	269	31	140
1500	120	55	206	44	85
1600	120	51	116	59	29
				DAILY TOTAL	1042

NO. DAYS: 31

AVG. OUTPUT = # DAYS × (% SUNSHINE + 0.1) × DAILY TOTAL
= 31(.44 + 0.1) 1042 = 17,443 BTU/SF MONTH

Solar Energy Collected

MONTH: JUNE

% SUNSHINE: 72

TILT: 42°

COLLECTOR EFFICIENCY CURVE EQUATION: $Y = 0.67 - 0.34X$

TIME	AVG. COLL. TEMP. $\left(\frac{T_I + T_O}{2}\right)$ °F	AMB. TEMP (T_A) °F	INSULATION (I) BTU/SF.HR	INCIDENT ANGLE	SOLAR ENERGY COLLECTED BTU/SF·HR
0800	145	70	122	67	—
0900	145	73	181	55	97
1000	145	76	227	45	129
1100	145	79	257	37	150
1200	145	82	267	34	157
1300	145	84	257	37	151
1400	145	88	227	45	133
1500	145	90	181	55	103
1600	145	89	122	67	—
				DAILY TOTAL	920

NO. DAYS: 30

AVG. OUTPUT = # DAYS × (% SUNSHINE + 0.1) × DAILY TOTAL

= 30(.72 + 0.1) 920 = 22,632 BTU/SF MONTH

APPENDIX VI

SOLAR ENERGY USED/HEAT PUMP ENERGY CONSUMPTION

TILT: 32°   COLLECTOR AREA: 100 SF

MONTH	BTU X 10⁵ HEATING LOAD	DWH LOAD	TOTAL LOAD	SOLAR ENERGY COLLECTED	SOLAR ENERGY USED	HEATING LOAD BALANCE	% PARTIC.	HEAT PUMP ENERGY CONSUM. KWH	DHW ENERGY CONSUM KWH
JAN	57.4	21.9	79.3	16.4	16.4	41	21	447	642
FEB	44.4	19.8	64.2	20.7	20.7	23.7	32	259	580
MAR	36.5	21.9	58.4	23.9	23.9	12.6	41	137	642
APR	13.1	21.2	34.3	26.5	26.5	—	77	—	229
MAY	2.6	21.9	24.5	28.2	24.5	—	100	—	—
JUN	—	21.2	21.2	28.9	21.2	—	100	—	—
JUL	—	21.9	21.9	30	21.9	—	100	—	—
AUG	—	21.9	21.9	31.8	21.9	—	100	—	—
SEP	2.6	21.2	23.8	29.8	23.8	—	100	—	—
OCT	15.7	21.9	37.6	29.4	29.4	—	78	—	240
NOV	36.5	21.2	57.7	19.5	19.5	17	34	186	621
DEC	52.2	21.9	74.1	14.7	14.7	37.5	20	409	642
TOTAL	261	257.9	518.9	299.8	264.4	131.8	51	1438	3596

TOTAL ENERGY CONSUMPTION = HEAT PUMP + DWH + SOLAR SYSTEM

= 1438 + 3596 + 1572

= 6606 KWH

SAVINGS = 10403 − 6606 = 3797 X 0.025 = $93

INCREMENTAL COST/ANNUAL SAVINGS

@ $. 0.0245/KWH 2135/93 = $22.9

@ $ 0.05 / KWH 2135/190 = $11.2

@ $ 0.10 / KWH 2135/380 = $5.6

SOLAR ENERGY USED / HEAT PUMP ENERGY CONSUMPTION

TILT: 32° COLLECTOR AREA: 150 SF

MONTH	BTU X 10^5						%	HEAT PUMP ENERGY CONSUM. KWH	DHW ENERGY CONSUM KWH
	HEATING LOAD	DWH LOAD	TOTAL LOAD	SOLAR ENERGY COLLECTED	SOLAR ENERGY USED	HEATING LOAD BALANCE	PARTIC.		
JAN	57.4	21.9	79.3	24.6	24.6	32.8	31	358	642
FEB	44.4	19.8	64.2	31.1	31.1	13.3	48	145	580
MAR	36.5	21.9	58.4	35.9	35.9	0.6	61	7	642
APR	13.1	21.2	34.3	39.8	34.3	–	100	–	–
MAY	2.6	21.9	24.5	42.3	24.5	–	100	–	–
JUN	–	21.2	21.2	43.4	21.2	–	100	–	–
JUL	–	21.9	21.9	45	21.9	–	100	–	–
AUG	–	21.9	21.9	47.7	21.9	–	100	–	–
SEP	2.6	21.2	23.8	44.7	23.8	–	100	–	–
OCT	15.7	21.9	37.6	44.1	37.6	–	100	–	–
NOV	36.5	21.2	57.7	29.3	29.3	7.2	51	79	621
DEC	52.2	21.9	74.1	22.1	22.1	30.1	30	328	642
TOTAL	26.1	257.9	518.9	450	328.2	84	73	917	3127

TOTAL ENERGY CONSUMPTION = HEAT PUMP + DWH + SOLAR SYSTEM
= 917 + 3127 + 1652
= 5696 KWH

SAVINGS = 10403 − 5696 ⇒ 4707 X 0.0245 = $115

INCREMENTAL COST/ANNUAL SAVINGS
@ $. 0.0245/KWH 3198/115 = $27.8
@ $ 0.05 / KWH 3198/235 = $13.6
@ $ 0.10 / KWH 3198/471 = $6.8

SOLAR ENERGY USED/HEAT PUMP ENERGY CONSUMPTION

TILT: 32° COLLECTOR AREA: 150 SF

REV "A" DOMESTIC HOT WATER LOAD GIVEN PRIORITY OVER HEATING LOAD

MONTH	BTU X 10^5 HEATING LOAD	DWH LOAD	TOTAL LOAD	SOLAR ENERGY COLLECTED	SOLAR ENERGY USED	HEATING LOAD BALANCE	% PARTIC.	HEAT PUMP ENERGY CONSUM. KWH	DHW ENERGY CONSUM KWH
JAN	57.4	21.9	79.3	18	18	57.4	23	627	114
FEB	44.4	19.8	64.2	23	23	41.2	36	450	0
MAR	36.5	21.9	58.4	29	29	29.4	50	321	0
APR	13.1	21.2	34.3	36	34.3	—	100	—	—
MAY	2.6	21.9	24.5	42.3	24.5	—	100	—	—
JUN	—	21.2	21.2	43.4	21.2	—	100	—	—
JUL	—	21.9	21.9	45	21.9	—	100	—	—
AUG	—	21.9	21.9	47.7	21.9	—	100	—	—
SEP	2.6	21.2	23.8	44.7	23.8	—	100	—	—
OCT	15.7	21.9	37.6	40	37.6	—	100	—	—
NOV	36.5	21.2	57.7	22	22	35.7	38	390	0
DEC	52.2	21.9	74.1	16	16	52.2	22	570	173
TOTAL	261	257.9	518.9	407	293	216	56	2358	287

TOTAL ENERGY CONSUMPTION = HEAT PUMP + DWH + SOLAR SYSTEM

= 2358 + 287 + 1652

= 4297 KWH

SAVINGS = 10,403 − 4297 ⇒ 6106 × 0.0245 = $150

INCREMENTAL COST/ANNUAL SAVINGS

@ $. 0.0245/KWH 3300/150 = $22

@ $ 0.05 / KWH 3300/305 = $11

@ $ 0.10 / KWH 3300/611 = $5.4

DISTRIBUTION LIST

HQ AFLC/DEMU - 1

HQ SAC/DEM - 1

HQ ADCOM/DEM - 1

HQ ATC/DEM - 1

HQ AU/LGD - 1

HQ AUL - 1

HQ MAC/DET - 2

HQ AAC/DEMC - 1

HQ TAC/DEM - 1

HQ USAFSS/DEM - 1

HQ USAFE/DEM - 1

HQ USAFA/DEER - 1

HQ AFRES/DE - 1

DET 1, HQ ADTC/SUL - 1

Chief, Naval Facilities - 1
Engineer Command
Attn: Chief Engineer
Department of the Navy
Washington, DC 20350

Civil Engineering Laboratory - 1
Energy Program Office
Code L80, NCBC
Port Hueneme, CA 93043

CERL - 1
P.O. Box 4005
Champaign, IL 61820

CRREL - 1
P.O. Box 282
Hanover, NH 03755

HQ AFSC/DEM - 1

HQ AFCS/DEM - 1

HQ USAF/LEE - 1

HQ USAF/LEEE - 1

HQ USAF/LEEM - 1

HQ USAF/LEEN - 10

AFRCE-ER - 1

AFRCE-WR - 1

AFRCE-CR - 1

AFIT/DE - 3

AFIT/LIBRARY - 1

NGB/DEM - 1

HQ USAF/LEYF - 1

DET 1, HQ ADTC/ECW - 5

Office, Chief of Engineers - 1
Attn: DARN-RD
Dept of the Army
Washington DC 20314

US Army/FESA - 1
Bldg 374
Ft. Belvoir, VA 22060

Dublin-Bloome Ass P.C. - 10
312 Park Road
West Hartford, CT 06119

Department of Energy - 1
Mail Stop 2221C
20 Mass. Ave. NW
Washington DC 20543